MARINE BOILERS

MARINE BOILER MANAGEMENT AND CONSTRUCTION

BEING A TREATISE ON

BOILER TROUBLES AND REPAIRS, CORROSION, FUELS AND HEAT
ON THE PROPERTIES OF IRON AND STEEL, ON BOILER MECHANICS
WORKSHOP PRACTICES AND BOILER DESIGN

BY

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PREFACE.

WHILE reading through these pages for the last time, and thereby completing a task which has proved a heavier one than could at first have been imagined, numerous passages have recalled to mind friendly discussions of which they are the outcome, or valuable hints and sometimes exhaustive criticisms from those friends to whom doubtful points were submitted. Should they find that their views have not in all cases been adopted, the text will doubtless also reveal to them the reasons why this could not be done; and to these friends I wish to convey my warmest thanks for the encouragement which their personal interest in this work has afforded me.

While collecting the material for this work a feeling that many problems yet remain to be solved has rarely been absent from my mind, more especially when the scientific side of a question was being inquired into. Not being in a position to satisfactorily discuss such problems, it seemed necessary at least to state them concisely, so that scientists might be induced to solve them for us. It would have been very easy to ignore such difficulties altogether; but this course would have been contrary to my purpose, which was to produce a work of a practical character. If it should be objected that just because of this object even the simplest mathematics ought to have been omitted, such critics should remember that true practice, unlike abstract science, is unscrupulous in the choice of means, and avails itself of the gratuitous labours of scholars and scientists as readily as it adopts the more costly experiences gained by repeated failures. Besides, as the object of science is truth, practical men, whose sole aim is success, dare not remain in the dark as regards its discoveries and deductions.

C. E. S.

GLASGOW: *August* 1893.

INTRODUCTION.

THE information contained in this work has been collected for the use of people interested in the manufacture and management of marine boilers, and it is hoped that the summaries of the experiences gained in one of the branches with which it deals will assist those whose attention has been confined more particularly to the other. Thus, for the information of manufacturers, it was necessary to discuss the troubles to be expected from the use of defective materials, as well as the dangers to which a boiler is exposed after it leaves their hands; while for steam users, descriptions as to the processes of construction, and scientific inquiries about corrosion, fuels, and similar subjects, had to be brought together.

In spite of a very exhaustive search, extending over several years, little information either about the management or about workshop practices could be found, most books and papers professing to deal therewith doing so only in vague terms. The Author had, therefore, to rely mainly on his own experiences when explaining the various practices and manipulations. Although it was impossible to enter into every minute detail, it is hoped that no points have been omitted to which attention should be drawn.

In writing the chapter on 'Mechanics' the Author also had to rely mainly on his own resources when trying to present such problems as occur in marine boilers in as simple and yet as comprehensive a form as possible. Special attention has there been paid to the relations existing between elastic stresses and those which make their appearance just before rupture takes place, whereby it is hoped that the term 'factor of safety' has acquired a more precise meaning than it at present possesses. A graphic method for resolving stresses, a method for estimating the shearing strengths of a material from torsion experiments, and a discussion on the irregular distribution of stresses in

riveted joints, are some of the subjects herein discussed, and about which little will be found in earlier books.

In the chapter on 'Corrosion' attention is drawn, amongst other matters, to the strange influence which apparently harmless salts exert on the harmful activity of weak acids; the action of air in feed-water and the much-debated question of galvanic currents in boilers has been treated in some detail.

While examining the numerous experiments on 'Heat Transmission,' and before Mr. Durston's excellent paper was read, the Author was confronted with the serious difficulty that nearly all these experiments are incomplete as regards certain essential points. Sometimes the heating value of the coal was not stated; sometimes the steam pressure, funnel or feed temperature, was forgotten, or the ashes not weighed; but by having brought together various experiments in one chapter a better idea can now be formed than has yet been possible about the resistance encountered by heat when it travels from the flame through the iron and scale into the water.

Somewhat similar remarks might be made on the subject of 'Strength of Materials,' for metallurgists are still unable to explain why a metal which can be stretched from 20 to 30 per cent. in a testing machine sometimes shows no plasticity, and cracks spontaneously when fitted in a boiler. Pains have been taken to collect all references to such influences as might cause trouble when using steel, and it is hoped that the Author's experience, both in engineering works and in numerous English and foreign steel works, have led him to touch upon everything that is essential.

The chapter on 'Fuels and Combustion' ought to be of value to those who are entrusted with the carrying out of accurate experiments on the performances of engines and boilers. Not only has it been explained there how necessary it is to know the heating value of the fuel used, and whether it has been properly burnt, but explanations have been added showing how to carry out the experiments, and as they are comparatively simple, sea-going engineers possessed of a knowledge of chemistry might easily inform themselves on questions such as how much heat and unconsumed fuel are escaping up the funnel, and how much priming water is carried over with the steam, about which points reliable information is still wanting.

For the convenience of draughtsmen the rules of the Board of Trade and of Lloyd's Register on the scantlings of boilers have all been placed near the last pages, and much trouble has been taken to make

the tables which are based on them not only reliable and comprehensive, but also compact. But as these rules are liable to occasional amendments, the tables may also have to be revised from time to time.

These rules and tables are preceded by a short chapter on 'Design,' in which such hints have been given as will, it is hoped, facilitate drawing-office work. It has not been thought advisable to reproduce any drawings of complete boilers, but a long list has been compiled of publications where they can be looked up, together with their principal dimensions, their performances, and the builders' names.

These and the other references to publications were found to be so numerous that it was necessary to abridge their titles to a few initial letters, which have been rearranged in alphabetical order in the following list, together with the necessary particulars for easily finding the required volume. It was at first intended to quote only such papers or books as had been printed in the English language, but many of these turned out to be translations, and it soon became evident that they could not always be relied upon. Not only were the misprints sometimes of the most serious nature, but in several instances it was found impossible to trace the original paper, because of the date, volume, or page being wrongly given. The plan had therefore to be adopted of mentioning only the original notice of experiments on investigation, no matter in what language they were first described.

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Am. J. S.	American Journal of Science. Newhaven, Conn.
Am. M. E.	Transactions of the American Society of Mechanical Engineers. New York.
Am. Min. E.	Transactions of the American Institute of Mining Engineers. New York.
Am. R. M. M. A.	Report of the Proceedings of the American Master Mechanics' Association. New York.
An. Ch.	Annales de Chimie. Paris.
An. Ch. Ph.	Annales de Chimie et Physique. Paris.
An. Génie	Annales du Génie Civil. Paris.
An. Ind.	Annales Industrielles. Paris.
An. Mines	Annales des Mines. Paris.
An. Pont. C.	Annales des Ponts et Chaussées. Paris.
Arch. I. V.	Zeitschrift des Architecten- und Ingenieur-Vereins. Hanover.
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MARINE BOILER MANAGEMENT

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CHAPTER I.

BOILER MANAGEMENT.

SUCCESSFUL management of a boiler consists in getting the desired amount of work out of it, and at the same time keeping the expenses as low as possible. To be able to do this, a boiler must above all things be well designed for its work; it should be handled with the proper amount of care, and any defects which may show themselves should be made good at once and their causes removed.

Lighting Fires.—The fires in the main boiler are usually lighted by throwing some burning fuel from the donkey boiler into the main furnaces; but if this cannot be done, a small fire is kindled on one of the grates and the others lighted from it. In either case the whole of the fire bars are first covered with coal, so as to restrict the draught to that point where the fuel is burning; gradually, as the fire increases, the hot fuel is raked back, igniting the remainder. If the coal is very dusty it may be necessary to cover the bars with old matting or with paper.

If steam has to be raised quickly, the fires are started in all the furnaces at once; but this should never be done unless there is steam in the donkey boiler, and appliances are available for producing artificial circulation. Even where this is the case, and steam has to be raised with the utmost possible despatch, it is safer to force only one boiler, or such a number as are sufficient to drive the engine; for in all cases of great hurry, things are likely to go wrong, and it has happened over and over again that circumferential seams of boiler bottoms have cracked when not sufficiently warmed during this period. The circulating appliances used to be hydrokineters; now it is customary to fit a donkey suction to each boiler bottom, by which means the cold water is drawn off there and reintroduced through the feed near the water level.[1] Where none of these arrangements are fitted, the heating of the boiler must progress slowly, taking about twelve hours. Many engineers start fires only in one furnace of each boiler, usually the lowest one, in the hope that the cold water, which remains undisturbed

[1] These pipes should be so fitted that water cannot accidentally be forced from one boiler into another if their pressures should differ.

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at the bottom, will get warmed. A better plan is to light fires in one of the wing furnaces, F, fig. 1, of each boiler, which sets up a natural circulation, the water rising on one side and descending on the other.

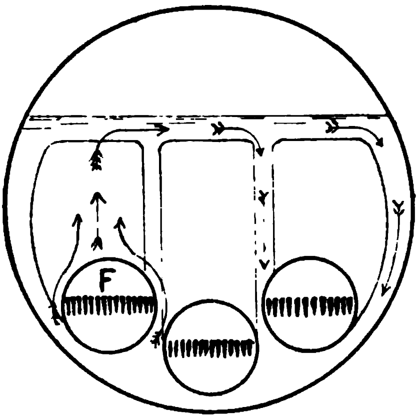


Fig. 1.

If steam must be raised quickly, a good plan is to fill the boilers to nearly full glass, to light all the fires at once, and to blow off all the cold water below the furnaces as soon as that above them is boiling. From fear of subsequent leakage, the safety valves are usually kept closed, and the result is that the heated air in the steam space shows a pressure on the gauge before steam has been generated, which is most misleading, because it all disappears as soon as the engines are started. This air pressure on the water surface also prevents the formation of steam bubbles when 212° F. has been reached, which would materially assist the circulation, by carrying water with them as they rise. It is, therefore, better to keep the main or safety valves full open till steam is blowing off freely, and only to close them when the boiler bottom has grown hot. If closed earlier, the chances are, even with slow firing, that the boiler bottoms remain cold long after the full working pressure has been reached, and boiler shells being constructed mainly to resist the steam pressure, are then sometimes unable to bear the additional straining to which they are subjected by the very serious differences of temperature between the upper two-thirds and lower one-third of their circumference. (See fig. 93, p. 53.)

Cracked Shell Plates.—This difference of temperature can amount to as much as 270° F., and, as iron expands about one-thousandth of its length between 32° F. and 212° F., the bottom of the boiler shell would tend to be about one-seven-hundredth of its length shorter than the top, which, in a double-ended boiler of 17½ feet length, would amount to more than ¼ inch. Of course the upper two-thirds of the boiler would be slightly compressed, but the amount of metal in the lower third being the smaller, suffers the severest stress, probably quite two-thirds of that which the difference of temperature would warrant. But if iron or steel is prevented from contracting one-thousandth of its length, which is the same thing as stretching the metal by that amount, a stress fo 13 tons per square inch is set up, and this, then, is approximately the extra stress which the lower third of the boiler shell has to resist.

As the percentage of strength of the circumferential joints is comparatively low, it is not to be wondered at that either the rivets shear or that the metal tears. In either case there is no immediate danger, for the leakage is slight and the water cold, at least at first. However, sometimes the solid plate cracks circumferentially, and then the rush of water is considerable; but it is only amongst iron boilers that this happens, which is doubtless due to the fact, that this material is decid-

edly weaker across the grain than with it. As might be expected, double-ended boilers are more often injured in this way than single-ended ones.

Those who wish to obtain numerical results on this subject should fit the following arrangement to the backs of a boiler, one near the water level and the other near the bottom, or wherever they think it most convenient. A short iron tube (fig. 2) about ¾ of an inch in diameter, and closed at one end, is screwed at an angle of about 30° into the back plate, and a little mercury or oil poured into it. When desired a thermometer can be inserted, and the temperature measured. If mercury is used the tubes must not be of brass or gun metal, otherwise they will be eaten away.

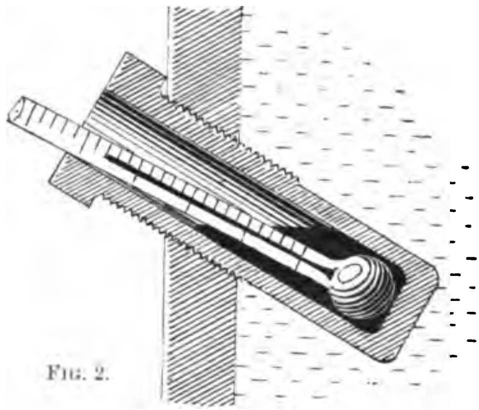


FIG. 2.

Stoking.—A perfect knowledge of stoking can only be gained by a practical experience, which does not fall to the lot of most engineers. It is not difficult to throw shovelfuls of coal through the comparatively small fire doors, even when they are 4 feet above the floors, but generally inexperienced hands will not be able to place the fuel on those parts of the grate where it is wanted, even when the conditions are favourable—and that is one of the chief secrets of good stoking, particularly with Welsh and similar coal, which may not be disturbed after it has been put on the grate. North-country coal has a tendency to cake, whereby the air passages are choked. While breaking up this fuel with a rake or slicer an excellent opportunity is afforded for levelling it.

While burning Welsh coal the case is different. The depressions, where the air meets with least resistance, are burnt away quickest, even to such an extent that the upper edge of the bars becomes exposed, admitting a damaging excess of air (fig. 3). On account of this short cut very much less air finds its way through the thick fuel, and the combustion is reduced there. Uncovered bars and thin fires are readily discovered on trials by holding an anemometer in each ashpit. On recoaling, the hollows will most likely be more than filled up ; but even if levelled, the combustion will be fiercest where the hot fuel was lying thickest, for there it is all aglow (fig. 4), and these are the parts which will now burn away quickest. Continued care is therefore necessary.

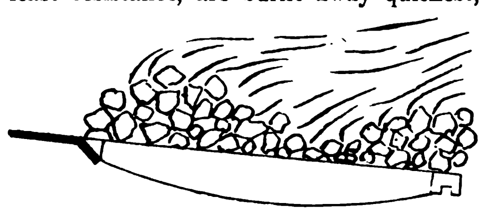


FIG. 3.

What is true of a single fire is true of a number, and unless all the fires, connected to one funnel, are kept equally thick the thin ones will burn away fastest, and if there are several stokers on one watch, he will have the lightest job who chokes his furnace with fuel. When steaming under strong forced draught with closed stoke holes, speed in coaling is of the utmost importance, so as not to keep the doors open too long, and it is customary to have a right- and left-handed stoker for each fire, with a leading man to direct them where to throw the fuel. In order to see better which parts of the grate are bare he should wear green-glassed spectacles.

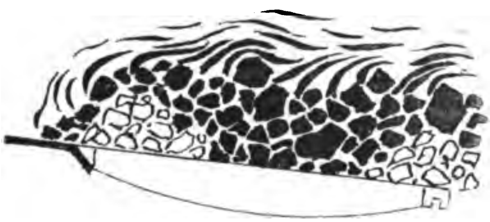


FIG. 4.

Thickness of Fires.—The question as to how thick the fires ought to be kept is a difficult one to answer. Less resistance will be offered to the passage of the air, and the combustion will be more rapid, the thinner they are. Thick fires, or rather heavy firing, offers the advantage that the furnace doors have to be opened less frequently, and the loss and injury due to inrushes of cold air are not so great; but such fires produce an enormous amount of volatile products, which are often so cool that they cannot ignite, as they should do, when mixed with the air which is admitted through the doors. This loss is, of course, greater with bituminous coal.

Under natural draught the fires should be kept about 6 and 8 ins. thick respectively for North-country and for Welsh coal. Under forced draught they are sometimes 12 ins. thick.

Smoke Consumption.—In order to consume the volatile gases some firemen coal first one side of a grate and then the other, half of the upper surface being thus always in a glowing condition. Another plan—but this can only be carried out with caking coal—is to throw the green coal on the front end of the grate, and to rake it back when it is well alight. The combustible gases which are at first given off are thoroughly mixed with the air which reaches them through the doors, and passing over the red-hot coal the mixture is bound to ignite.

This plan has the advantage of allowing very long grates to be used. However, the ashes and clinker will be driven to the back ends, against the firebrick, whence it is most inconvenient to remove them. With dirty coals this trouble is so serious that firemen prefer to throw them as far back as possible, and to rake them forward when partly consumed, accumulating the ashes at the front end. This is an uneconomical proceeding.

Air Admission above the Grate.— It is conceded on all hands that a certain amount of air must be admitted above the grate, and that it is difficult to determine how much; but it will not be out of place to illustrate the subject by drawing attention to the behaviour of the flame of an ordinary paraffin lamp. If the wick is turned too low disagreeable smells are produced, filling the room in a very few minutes: if too high, other smells and much soot are produced. In the one case we have incomplete combustion, due to the cooling action of an excess

of air ; in the other case too little was supplied. Smoke and smells are emitted from funnels, showing that here too unconsumed gases are escaping ; for carbonic acid, as well as steam, is without odour.

With present arrangements it is almost impossible to regulate the air supply, or at any rate to fix it, so that the ratio of the excess to the total shall have a definite value, and consequently we find in practice that it varies from 25 % to 200 %. (See pp. 71 and 78.)

It is clear that when the fires are thick less air passes through them less fuel is consumed, and relatively more air is drawn in through the doors. With thin fires the draught through them is stronger, more fuel is burnt, and less excess air drawn in. Now, although the products from thick fires contain more inflammable gases than those from thin ones, and consequently require more excess air than these, it is difficult to believe that they will receive just their correct share, whatever the sizes and number of holes in the doors, whatever the state of the grate, and whatever the intensity of the draught or the quality of the coal ; and it is only reasonable to assume that the best results will be obtained—

I. If no air is admitted through the doors when the fires are very thin and all aglow.

II. If much air is admitted immediately after coaling and when the fires are thick.

III. If more air is admitted with North-country than with Welsh coal.

IV. If less air is admitted as the fires get dirty and the combustion reduced.

But even with the most correct proportions the combustion will not be perfect, if for no other reason than that the flame comes in contact with the boiler plates, and gets cooled, before it is burnt out.

As regards the excess air, it is perhaps of more importance to decide where to admit it than how much to admit, provided it be enough. If it were possible to keep the fires very thin, it might be well to admit all the air through the bars, and none above the fire ; but this is impossible. To admit air through the furnace doors stimulates combustion at this point, so that, before mixing with the distilled furnace gases, it has already been robbed of much of its oxygen. Just after coaling this is not the case ; but being even colder than the products of distillation, it will cool instead of igniting them. As already mentioned, this result can be evaded by throwing the green coal only on the front ends ; but another plan is to have much brickwork at or behind the bridge, which soon grows very hot, and the mixed gases ignite on coming in contact with it, and burn in the combustion chamber. Unfortunately the distance to the tubes being short, the flame is immediately extinguished on entering them.

The plans have also been tried of admitting air at the bridge, or through a tube passing from the shell direct into the combustion chamber; but both are open to many objections, not the least being that the passages get choked and are certainly not under control, and can neither be cleaned nor closed when desired.

With those systems of forced draught in which the ashpits are closed, the funnel damper can be set so as to retard the draught above the bars to such an extent that just the right amount of air is admitted, even when the doors are wide open.

Flames.—Interesting experiments have been made to show that on mountain tops, where the air pressure is much reduced, and also in partial vacuums and bad atmospheres, ordinary flames lose their luminosity, while under high pressures they grow smoky, and even the hydrogen and carbonic oxide flames grow luminous. From this it has been argued that draught-retarders and the closing of the funnel damper, when using forced draught, will improve combustion. But those very experiments (E. Frankland, 1877, p. 876) go far to show that the combustion is not accelerated ; besides, the increase of pressure attainable by these means is so slight in comparison with the atmospheric pressure variations, that the influence would have shown itself before now if it were true that more heat can be got out of coal with a high than a low atmospheric pressure.

Some interesting facts as to the igniting temperatures of various substances will be found in the chapter on 'Fuels and Combustion.' There will also be found a good deal of information on the heating-power of fuels, and on various methods for measuring it, as well as for determining, from the funnel gases, whether the combustion is perfect or not. Some of these tests are so simple that they can easily be carried out at sea.

Furnace Doors.—As regards economy of fuel, furnace doors unquestionably rank amongst the most important attachments to a boiler, and innumerable are the patents in connection with them ; but the very fact of numberless ideas having been published makes it impossible to deal with any of them as exhaustively as might be wished, and only those will be mentioned which have found their way into the stokehold.

The door may consist of a single plate, fitted with hinges and a latch, and perforated, chiefly at the upper edge, or fitted with a gridiron or other contrivance for regulating the admission of air. The objection to this arrangement is that, on account of the direct radiation from the fuel, the door very soon gets excessively hot and warps, and even cracks. The presence of an air regulator is an advantage, if properly used, but the very reverse if its action is not understood (see p. 5). Some doors are hinged, so that they can be kept partly open. In order to protect the door from the heat, an inside screen should be fitted, which can be renewed when burnt away ; for, as it cannot be kept as cool as the outer one, it suffers more severely. A simple arrangement is to rivet a plate *a*, fig. 5, having a number of holes, to a solid one, *b*. The air then enters at the circumference and passes through the various holes, as shown by the arrows.

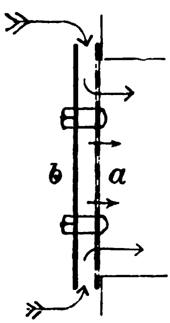


FIG. 5.

In other cases the internal plate is so fitted that the current of air is directed either upwards or downwards, according to the views of the respective engineers. (See figs. 6, 7.) An idea seems to prevail that by leading the air through complicated passages it collects heat, thereby facilitating combustion ; but this warming is so slight that it does not justify expensive arrangements. Wide dead plates in combination with doors which admit air only at the top, keep the latter cool. As already mentioned, hinges are sometimes constructed so as to keep

the doors partly open; and certainly they should all be arranged so that they will keep quite open when coaling, particularly during rough weather. Some of the contrivances used for this purpose are very simple, as well as fairly efficient. All of them should be strong, and capable of being worked in the easiest possible manner, for a fireman's chief tool is his shovel. Besides, on account of the heat, anything belonging to a fire door cannot well be touched by hand. One way of keeping the doors open is to balance them either by weights or springs, but with most of these arrangements the ashpit gets closed during firing, whereby that part of the air supply which passes through the fuel is lessened, and that which passes through the door increased, doing harm where it chills the various plates.

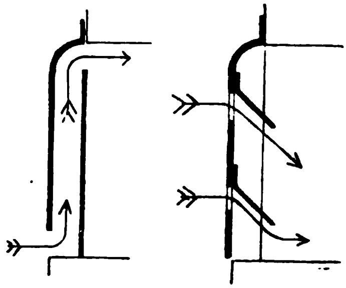


FIG. 6. FIG. 7.

Door Frames suffer in the same way as the doors to which they are attached, unless they too are properly protected from the heat, either by baffle plates with air admissions at the back, or more generally by firebricks. It is never good to make the door frame in one piece, as it is sure to crack. With large furnaces, of 40 ins. diameter and above, two doors are sometimes fitted. Attempts have also been made to fit feed heaters at these points, but evidently without success.

Fire Bars.—It cannot be said that this subject has been neglected by inventors, for the patents in connection with it are innumerable; but it is certainly very unsatisfactory that, after trying various novelties, engineers always fall back on the old pattern, viz. length $2\frac{1}{2}$ to 4 ft., air spaces $\frac{1}{2}$ to $\frac{3}{4}$ in., depth $3\frac{1}{2}$ to 5 ins., and thickness $\frac{3}{4}$ to 1 in. at top, tapering 1 to $1\frac{1}{2}$ in. per foot of depth. Even for forced draught, if not excessive, the above dimensions give good results, although the Admiralty, who use wrought iron or steel instead of cast iron, make the bars about $\frac{3}{8}$ in. thick, while the air spaces are reduced to $\frac{3}{8}$ and even $\frac{1}{4}$ in.

It has been argued that by reducing the upper surface of the bars a smaller area is exposed to the heat of the fire, and that such bars will keep cooler; but a glance under the grate of a furnace is sufficient to convince anybody that as much if not more heat is received by radiation by the sides of the bars as by their upper surfaces. A closer examination will also show that each bar is surrounded by a visible layer of trembling hot air, which is moving upwards and seems to be about $\frac{1}{16}$ in. thick. It is the heating of this thin film of air which keeps the bars cool. These facts might lead to the conclusion that a distinct advantage would be gained by reducing the thickness of each bar, and fitting more of them, because thereby relatively more cooling surface is obtained. But if, as seems necessary, the air spaces are left as wide as before, each bar will receive an extra amount of heat, so that the thin bars will probably grow quite as hot as the thick ones, and if that is the case they are at a great disadvantage, for being thin they are sure to get bent sideways. In fact, the only way to use them is to pack them tightly into the furnace, so that they can neither bend nor twist.

Where great trouble is experienced water in trays is placed in the ashpits. This seems capable of abstracting sufficient heat, but the air channels under the bars are seriously reduced, and salt water in the furnaces is not a desirable object, as it causes a lot of corrosion. On forced-draught trial trips it is often necessary to keep hoses playing sea water into the ashpits. No doubt the accumulation of ashes and red-hot small coal in the ashpits keeps the bars hotter than they should be, and, as they also seriously interfere with the draught, it would be a great advantage if means could be devised for removing them.

Another plan for keeping the bars cool is to make them deeper. Heat travels so very quickly in metals that the small extra distance which it has to go before reaching the lower edge hardly affects the result, which, roughly stated, is, that the temperature of the bars is inversely proportional to their depths, or more correctly to their exposed surface, that their rigidity (horizontally) is proportional to their depth and to the square of their thicknesses. The horizontal deflection of a bar, to which a definite curvature has been given, is proportional to the square of its length. These views lead to the following formulæ, with whose help the small table has been compiled. If not numerically correct, it can at least be used for making comparisons.

I. For a given coal consumption, and for a given length of fire bar, the sum of the sectional areas of the bars contained within 12 ins. of the furnace diameter should be a constant value.

$$n.t.d = C_1.$$

II. For a given coal consumption the sum of the sectional areas of the bars contained within 12 ins. of the furnace diameter should be proportional to their lengths.

$$n.t.d = C_2.l.$$

III. For equal lengths of fire bars the square of their depths should be proportional to the weight of fuel burnt per square foot per hour.

$$d^2 = C_3.Q.$$

In the above formulæ the various letters have the following meanings :—

n stands for number of fire bars per foot of furnace diameter.
t stands for thickness of fire bars at their upper edges.
d stands for depth of fire bars at their centres.
l stands for length of fire bars.
C_1, C_2, C_3 are constants.
Q is consumption of coal per square foot per hour.

Values of the Products n.t.d. (This is the smallest permissible Sum of the Sectional Areas in Square Inches of Cast-Iron Fire Bars which must be contained within One Foot of Furnace Diameter.)

Coal Consumption per Sq. Ft. of Grate per Hour	20 lbs.	40 lbs.	80 lbs.	100 lbs.
Length of fire bars = 6 ft.	72	100	—	—
" " 5 ft.	40	55	80	100
" " 4 ft.	32	45	64	80
" " 3 ft.	24	34	48	60
" " 2 ft. 6 ins.	20	28	40	50
" " 2 ft.	16	24	32	40
" " 1 ft. 6 ins.	12	17	24	30

With bars whose air spaces are one-half of their thickness—i.e. 4 ins. per foot of furnace front—the depths would be found by dividing any of the numbers in the table by 8 ins. Thus with bars 2 ft. 6 ins. long, burning 40 lbs. per hour, the number is 28, and the minimum depths of such bars would be $3\frac{1}{2}$ ins., while the thickness might be made $\frac{3}{4}$ in., with $\frac{3}{8}$-in. air spaces, or $\frac{1}{2}$ in. with $\frac{1}{4}$-in. air spaces.

In cases of forced draught these two values are very often equal—say, $\frac{3}{8}$-in. bars and $\frac{3}{8}$-in. air spaces. To obtain the depth in their case, the number in the table would have to be divided by 6 ins., so that with a consumption of 40 lbs. the 2 ft. 6 in. bars would have to be $4\frac{2}{3}$ ins. deep, and 18-in. bars would have to be 3 ins. deep.

Naturally these values are only approximate, and depend very much on the fuel, but they may serve as a guide when making alterations.

Furnace Diameters.—It will be noticed that as soon as the usual practice is departed from, either by increasing the length of bars or the coal consumption, then their depth grows so great that it seriously interferes with the draught. This influence is particularly noticeable in boilers with small furnaces. Compare, for instance, two flues, the one being 33 ins. in diameter and the other 48 ins. If, as is usual, the lines of the dead plates pass through their centres, then the sectional areas below and above these lines are 3 sq. ft. in the one furnace and 6·3 sq. ft. in the other. With fire bars which are 3 ins. deep, the ashpit areas are reduced to 2·3 and 5·3 sq. ft. respectively, or ·85 and 1·33 sq. ft. per foot of furnace front. Under ordinary conditions this means that in the one case the air entering the furnace has to travel with a velocity of 12 ft. per second, in the other case its velocity is only $7\frac{1}{2}$ ft., and the resistances would be as 3 to 1. An extra inch added to the depth of the bars would increase the one resistance 25 %, and the other only about 6 %, so that in cases where the performance is low there is much less chance of efficient alterations if the furnaces are small than if they are large.

The sum of the air spaces between the fire bars usually amounts to 33 % of the width of the grate, so that if the length is 5 ft. we have $1\frac{2}{3}$ sq. ft. of air passage for every foot of furnace front. This being about twice as large as the ashpit area of the small furnace, little improvement would be effected in the draught and combustion of the small furnace by giving wider air spaces, for the depth of the bars would also have to be increased, whereas in the large furnace, where the ashpit area is sufficient, the alteration might increase the combustion.

On trial trips, and whenever it is desired to obtain the highest performances, the ashpits should be kept clear of ashes at all times, for every inch of piled-up material restricts the draught. Ash ejectors might be used with advantage.

With forced draught the case is different. Except on trial trips, where the air pressure is limited, there is practically no restriction as to the pressure which may be applied, and all the above reasons for allowing large air spaces fall to the ground, and there seem to be no objections against reducing them to the very narrowest limits. In fact, very good results are said to have been obtained in some foreign vessels where the air spaces have been reduced to $\frac{1}{25}$ in. and a high-pressure blast introduced into the ashpits. The necessity for opening the

lower doors to remove the ashes does not exist, for none can fall through; they are all fused and form clinker on the bars. Of course such bars could not be used for natural draught; but natural draught in the Navy is simply another name for a very low fan pressure, which could be increased if desired.

The following advantages should not be overlooked:—By reducing the air spaces till they are mere slits, the chief resistance to the air passage is found at these points and not in the fuel, and the draught

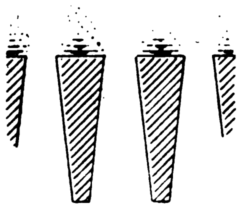


FIG. 8.

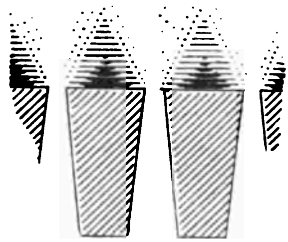


FIG. 9.

will hardly be affected by considerable variations of thickness of the fires. The chief combustion takes place just over the slits, and little, if any, over the centre of the bars, which remain covered with comparatively cool fuel or ashes, and are therefore not exposed to so much heat as the narrower bars with wide air spaces. An attempt to indicate this difference is shown in figs. 8 and 9.

The Burning of Fire Bars is often the consequence of irregularities in the upper surface of the grate. It is but natural that if several air spaces are blocked (fig. 10), either by clinker or when the bars are bent, then the air can only cool one side of such bars, and these must grow hotter than the others. If, in addition, fuel should get wedged into the wide space, nothing will prevent the corners of the bar from burning, and as these waste away the fuel sinks lower and lower, and it is only a question of time as to when it will have effected the destruction of the bars; naturally the adjoining ones, which are then exposed to the same action, will suffer in the same way.

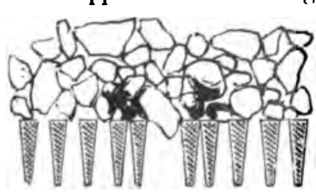


FIG. 10.

An examination of the grates of a boiler which has been worked hard will show that the bars have all sagged more or less, and that some are bent sideways, but that the upper edges are all uniformly wasted away, presenting a regular surface. On removing and then replacing them this condition is entirely altered. Possibly bars from the sides have got placed near the centre, and *vice versâ*, and a section across the grate might have the appearance of fig. 11, with the inevitable result that the tops of the projecting bars would be quickly burnt away. Possibly this may lead to the worst ones falling out, but at any rate the bars will generally be more quickly wasted if placed together in this manner

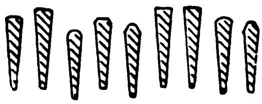


FIG. 11.

than if kept level. Where the grate is made up of two lengths it is always best to place any new bars at the back end.

Ashes and Clinker.—Another great trouble, and one which, like all others, is aggravated under the conditions of forced draught, is the accumulation of clinker on the grate. Each ton of coal contains from ¾ cwt. to 3 cwt. of ashes, according to the quality, and although a portion drops into the ashpit, a considerable part remains on the grate. It also depends upon the nature of the fire and on the composition of the ashes whether much—say ⅔—are blown up the funnel, or whether only a small quantity—say 25 %—are disposed of in this way. If the fires are dull and the ashes refractory the latter remain dust and are carried away; but if they melt easily, or if the fire is a fierce one, they form clinker, which remains.

The following minerals are found in ashes of coal:—

Silicic acid	from	2 %	to	60 %
Calcic oxide	,,	1	,,	20
Ferric ,,	,,	15	,,	75
Alumina	,,	2	,,	40

The most refractory ashes are those containing the smallest quantities of silicic acid, and the most fusible those where it amounts to about 50 %, with 25 % calcic oxide. If the alkalies were absent the ashes would remain dust under the most trying circumstances. Salt increases the fusibility of clinker, and the hydrochloric acid, which it loses while melting, is a very active corrosive agent and injurious to the boiler plates. Sea water should, therefore, not be mixed with the coal, and leakages in the furnaces and combustion chambers should be prevented as much as possible.

Two qualities of coal, whose respective ashes are fairly refractory, will occasionally produce very fusible clinker, if used together. In such cases the one probably contains an excess of silica, and the other an excess of the other ingredients. The melting away of the firebricks of the bridges is sometimes due to the great heat, but more often to an excess of the alkalies in the clinker. They readily attack the silicic acid, which is the chief constituent of the firebricks. Basic bricks would not suffer under these circumstances, but cannot be used at sea, as they readily absorb moisture when out of use, and then fall to pieces.

Cinders.—When the ashes, which have not been blown away, melt, they trickle down the sides of the partly consumed small pieces of coke, and by glazing them prevent their quick combustion. This may be a slight advantage, as the point of highest temperature of the fire (see fig. 105, p. 74) is thereby raised a little above the grate surface. When these small pieces of fuel ultimately find themselves over an air space between two bars, they drop through and are wasted; those which, at the end of their downward motion, arrive on the top of a bar are consumed there; therefore if a sufficient amount of forced draught is available it would be more economical to reduce the air spaces to the smallest possible dimensions.

Clinker.—The slag which had adhered to the small coal is now resting on the fire bars, and congeals, or at any rate its lower part, as

well as its extremities where it is in contact with the comparatively cool air.

As more and more slag is added, troughs of thick slag are formed, whose edges project over the air spaces and gradually close them (fig. 12), seriously interfering with the draught and combustion. With worn bars (rounded tops) the closing is effected more quickly, as the slag trickles down the sides (fig. 13); and the very fact that this happens is

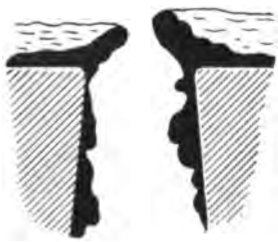


FIG. 12.

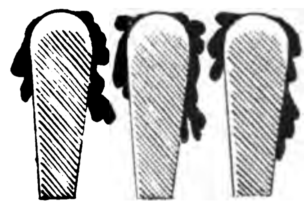


FIG. 13.

a sign that the bars are equally hot; and this is not to be wondered at, for the slag is sure to be very thin on the top, affording no protection from the heat, and is very thick at the sides, preventing the cold air from cooling the bars.

The necessity for guarding against this trouble has in America been the cause of the adoption of fire bars with hollow tops (fig. 14). The channels are soon filled with ashes or clinker, and it is only their two edges which absorb heat from the fire. It is stated that these fire bars, which are easily cast if the pattern is made in two halves, have a longer life than the ordinary ones, and also that clinker does not adhere to them so firmly.

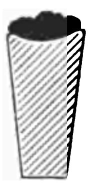


FIG. 14.

When fires have been burning for twenty-four hours they will be thoroughly dirty, the weight of mineral matter resting on one square foot of grate surface amounting to from 10 to 40 lbs., or from $\frac{3}{4}$ to 3 ins. in thickness. Of course the air spaces can then only be kept open by frequently using the slicer and pricker (figs. 17 and 19), but much can be done to break up the slag, which adheres to the bars, by moving them as indicated in fig. 15; this is only possible if they are very loosely packed, and it is customary to fit one bar less than the number which might possibly be got into a furnace diameter. They should be supported at the highest possible point, so that the air spaces at the top are not altered, but only the angles.

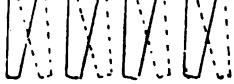


FIG. 15.

In order to allow for longitudinal expansion of the fire bars, one end, where they rest on the dead plate, should be made slanting. If there are two lengths, the other slant should rest on the bridge plate (fig. 16). Any bodily motion is prevented by notching the other end of each bar where it rests on the cross bar.

If the lengths exceed 3 feet, small distance pieces, A, are cast on at the centres; they prevent excessive distortions. The bars are also made

decidedly taper, 1 in. to 1½ in. per foot of depth, so that the fuel cannot be jammed between them, which would certainly happen if they

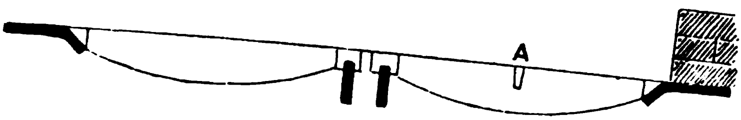


FIG. 16.

were parallel and loosely fitted. The navy fire bars, which are made parallel, have therefore to be packed tight.

Replacing Fire Bars.—In spite of all these precautions it will happen that fire bars burn, or under forced draught a whole grate may sometimes drop into the ashpit ; but this only happens when a large amount of red-hot ashes have accumulated there. Nothing can then be done but to clear out the furnace and rebuild the grate. When only one or two bars have dropped out, these can be replaced by new ones without drawing the fires. At the front end this is an easy matter, and some men are even sufficiently skilled to be able to throw a fire bar into its right position at the back of a grate ; but the safer plan is to tie it to a slicer, and then to place it in position. The yarn will very soon be burnt, and the slicer can then be withdrawn. Wrought-iron and steel fire bars, with their high melting temperatures, are said to last longer than cast-iron ones, but it is at present difficult to obtain them of a taper section, as very few rolling mills produce it, so that a large spare stock would have to be carried. Secondly, it is difficult, or rather expensive, to thicken the ends, though it would seem that with a suitable die and a steam hammer the problem could be readily solved. The plan has been successfully tried of notching the dead plate, bridge plate, and the cross bars, and using fire bars without thickened ends. In this case the air spaces cannot be kept equal at the upper edges, unless all parts are firmly fixed, which has already been shown to lead to trouble.

Cleaning Fires is one of those operations which, although necessary for the working, may cause a good deal of damage. Leakages in the combustion chambers are often attributed to the rush of cold air while the door is open and the grate bare. To close the funnel damper seriously interferes with the generation of steam from the other furnaces, and some ships have therefore been fitted with separate dampers over each nest of tubes (see fig. 30, p. 18), and the results are said to be satisfactory.

Another plan is to clean only half a fire at a time : one side of the fire is allowed to burn down, the clinker and ashes are raked out, and the bare bars covered with green coal, which is ignited by the fire of the other side, which, after a time, is treated in the same way. On account of the double operation firemen are not fond of this plan.

Another plan, which, to a certain extent, is indulged in by every fireman when burning caking coal, is to draw the clinker forward with a heavy rake (fig. 18) about one or two feet between each firing, taking particular care to throw the green coal on those parts of the grate which have been bared. The clinker should not be allowed to get cool,

otherwise it will be difficult to loosen it from the bars, which is done with the slicer (fig. 17).

Removal of Ashes.—Having got the ashes out of the furnaces, they naturally fall against the boiler, and are there slacked with sea water.

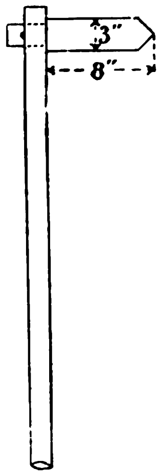


FIG. 17.

FIG. 18.

FIG. 19.

The noxious vapours which escape are an indication that corrosive acids are being generated, and it is only too well established that the most serious wasting to which the external parts of a boiler are subjected takes place here. Far better arrangements than those usually adopted should be made for preventing the ashes from resting against the boiler plates, or from getting into the narrow crevices which are found here. Besides cementing the inside of the end seams of boilers, it would not be amiss to cement the outsides as well where this corrosive action may be expected. When guard plates are fitted, no spaces should be allowed where ashes and moisture could find a lodgment.

Occasional efforts have been made to remove the ashes out of the stokehold by means of ejectors, and although successful for this purpose they quickly corrode, but other mechanical contrivances might be tried.

The amount of ashes to be dealt with in twenty-four hours can be roughly estimated as equal to the amount of mineral matter in the coal burnt during that time.

Mechanical Stokers. It is unnecessary to make any remarks about mechanical stokers, as everyone that has yet been tried at sea has been given up ; but for those who wish to study the subject the following references may be of value :—

J. Daglish, 'M. E.,' 1868, p. 155, mentions Stanley's and Vicar's patents. J. F. Spencer, 'C. E.,' 1891, v. civ. p. 54, mentions about a dozen different systems. F. Colyer, 1886, also mentions several.

Banked Fires.—An interesting table, containing the amount of coal consumed under banked fires, will be found in C. Busley's work, 1883, p. 157, from which it appears that from two to three cwt. per furnace

are consumed in twenty-four hours, and that the quantity required to raise steam is slightly less than this amount. He does not mention whether the fires were shoved back or drawn forward. The former is the more correct method, because it chokes the bridges, and the little air which passes there is hot; but, as the dampers can never be closed sufficiently, more heat would be developed than is required to maintain steam, and the more general practice is to draw the fires forward, and to leave the bars bare at the back. Much cold air is thus admitted, which keeps down the steam and reduces the draught. Of course the boilers suffer injury under both treatments, but particularly under the last-mentioned one.

When the engines have to be stopped without previous warning, the dampers are at once closed and the smoke-box doors opened; or (but this should not be allowed) all the furnace doors are thrown open. This is certainly a very effective means of reducing the heat of the fires, but the cold inrush of air may easily cause leakage. Should the steam still be rising fast the supplementary feed should be turned on before the safety valves lift.

Sweeping Tubes.—On account of the soot, which adheres to the tops and sides of the tubes, and the ashes, which settle along their bottoms, it is necessary to sweep them about once a week. Wire brushes (fig. 20) are used for the soot and ashes, and the split scrapers (fig. 21) to remove salt. The latter tool is of little use at sea, because a tube which is once plugged up with salt, closes up again after half an hour's steaming, and need therefore not have been cleaned. The brush should be strong enough to remove all the dust. These tools are hinged at their centres for convenience of handling. The sweepings of each tube naturally fall into the combustion chamber, but the draught at once carries the greater part into the adjoining tubes and up the funnel. In order not to get smothered the firemen keep the damper open, and very light fires on the grates, just sufficient to produce a draught. Except in large steamers, or in vessels fitted with several funnels, it is usual to clean the tubes of all the boilers at one operation, the whole watch being sent below for the purpose; for, as the draught is nearly non-existing during this time, little steam can be generated, and the sooner the operation is completed the better.

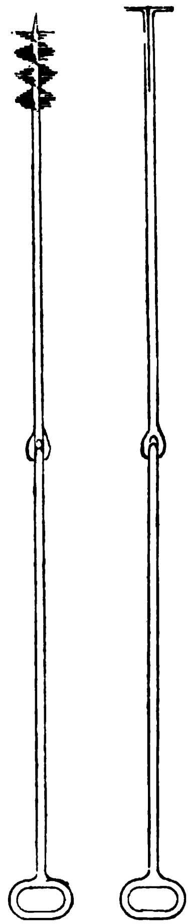


FIG. 20. FIG. 21.

Furnace Bridges.—The object of fitting bridges in furnaces is partly to keep the fuel from falling into the combustion chambers, and partly to reduce the air channel, so that, by imparting a momentarily high velocity to the flame, a thorough mixture of its gases is effected, and the combustion completed. The same result is aimed at in Mr. Holt's

boiler (fig. 22), in which the bottom of the combustion chamber is raised above the centre of the furnace, and the two are connected by a comparatively small air tube. One disadvantage of this arrangement is,

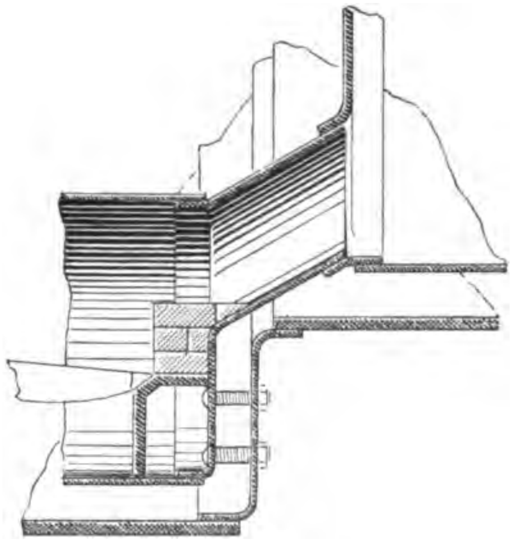


FIG. 22.

that no alteration can be made in the size of this air passage, and if the draught is insufficient to overcome the various resistances, the consumption cannot be increased. Brick fire-bridges are arranged as shown in figs. 23–26. The space at the back of the bridge shown in fig. 23 is usually quite full of ashes, even though a door is fitted at the bottom by which they could be removed.

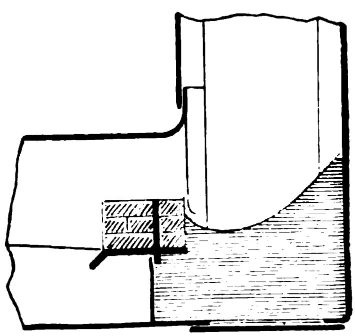


FIG. 23.

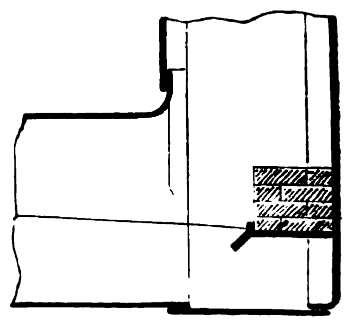


FIG. 24.

The heat which the firebricks collect may occasionally do a considerable amount of harm to the plates against which they rest if, as sometimes happens, the water is let out of the boiler before it is cold.

Viewed from the front, the bricks are generally placed horizontally; but where much trouble is experienced by their falling out of place,

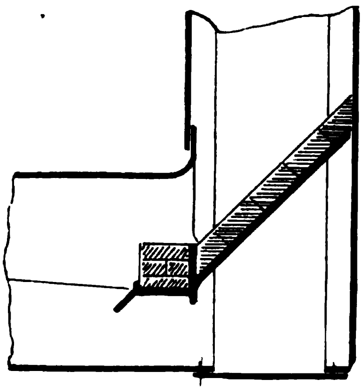


FIG. 25.

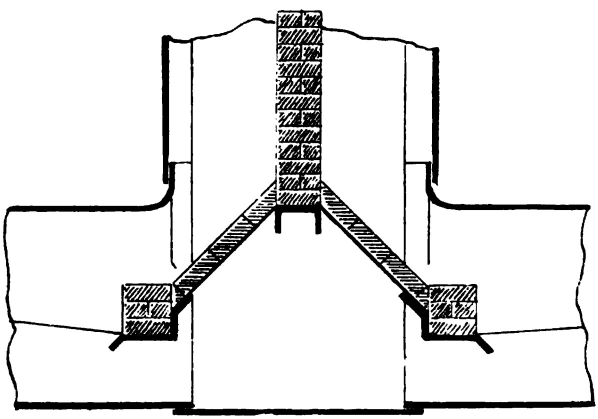


FIG. 26.

which happens when clinkers are allowed to adhere to them and have to be broken, or when the fires are forced, or when the furnaces are large, the bricks are sometimes arranged in the form of an inverted arch (fig. 27). In double-ended boilers, with through combustion chambers, the same plan has been tried on the central partition wall; but no advantage seems to have been gained, and where it is desired to carry the wall high up, it is safer to bolt two angle irons down the sides of the combustion chamber.

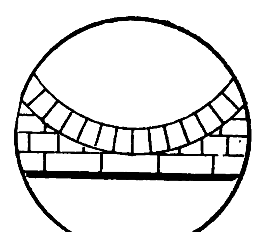


FIG. 27.

In cases where the furnace saddles have been cropped, or where the tube plate has been flanged to meet the furnace, exposing the back end seam to the direct action of the flames, it is sometimes necessary

to protect it by a firebrick arch (fig. 28). Here, too, it is an advantage if the bridge is curved, as it gives a better support to the arch. (See fig. 29.)

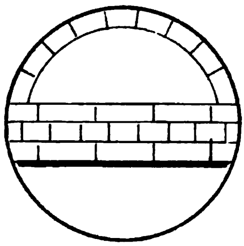


FIG. 28.

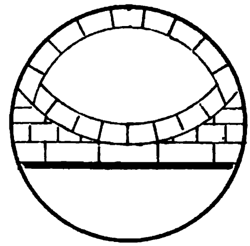


FIG. 29.

The Influence of Heat and Cold on Boiler Plates.—In the chapter on 'Strength of Materials' will be found information on the effect of repeated coolings and heatings of metals, which seems to show that this treatment can make iron and steel brittle. It also seems to be a fact that it tends to alter the shape of structures. For both reasons it is desirable to guard against such changes as much as possible, which means that the periods during which the fire doors are kept open should be few and short.

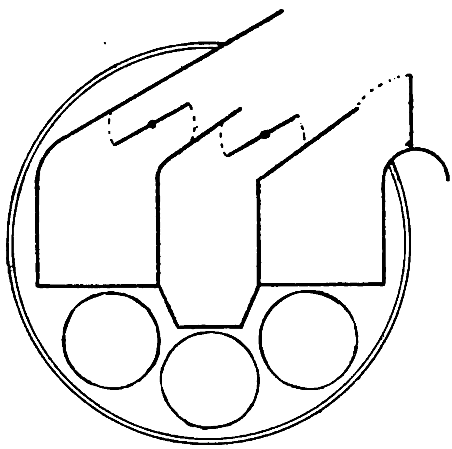


FIG. 30.

Dampers.—An alternative is to have a damper fitted in the smoke box for each combustion chamber, and while opening any fire door the respective damper could be partially closed. They can be arranged as in fig. 30. Something of this sort, but stronger, is desirable with forced draught in closed stokeholds, for the pressure is there much greater than with natural draught, and an enormous amount of air escapes when the doors are opened. But there are also strong objections against fitting any dampers in these cases.

If the forced draught is used with closed ashpits, dampers are also necessary, for the suction above the bars is not always equal to the pressure which is transmitted from below. In fact, when firing, it is customary to restrict the blast, because otherwise so much gas is generated that the funnel draught is unable to carry it off, and the fumes would enter the stokehold.

Flames in Stokeholds.—The most serious mishaps may be caused if steam is suddenly generated in the combustion chambers, which is a powerful reason why fusible plugs should not be fitted. With some boilers in the Navy sad loss of life has several times been caused by

sudden leakages of the back ends of the tubes. It has been suggested that this happened with H.M.S. 'Barracouta,' but a similar effect might be produced by stopping the fan in one of the stokeholds, the flames being at once driven into it over the bridge by the fan in the other stokehold, while the funnel is unable to draw them away. The difference of pressure would no doubt be increased by men anxious to render assistance blocking the stokehold passages. Similar accidents have also happened in single-ended and in navy-type boilers, in which this action is impossible.

Water Gauges.—Of the various boiler mountings, one of the most important is the gauge glass, and it ought always to be in good working order, and capable of being tested at any time.

In some cases it is fitted direct to the boiler plating; but this is not a good plan, for then the scum cannot be kept out of the glass, and the oscillations due to the rolling of the vessel and to irregularities of ebullition make it difficult to obtain good readings.

The usual plan is to connect the gauge glass to the top and bottom of the boiler by means of long pipes, the one reaching well up into the steam space, and the lower one into the water, where little circulation is expected to be found. These pipes can be fitted either externally or internally. The latter arrangement is not always possible, on account of the smoke box, and some engineers object to it *in toto*, on the principle that no copper pipes should be fitted inside a boiler. From the following table it will also be seen that, if not often blown through, gauges with long connections may indicate a different water level than actually exists in the boiler. When fitted externally these pipes have the effect of cooling the water contained in them, and as its density is thereby very much increased, the indicated water level is lower than that in the boiler. Matters are still further complicated by the accumulation of distilled water in the upper parts of the connecting pipe, which, if there is much salt in the boiler, has the effect of making the water level appear higher than it really is. The possible differences caused by a connecting pipe 10 ft. long are shown in the following tables, which contain the corrections which should be applied to the water-gauge readings under the various conditions.

Hot Water in Boiler. Cold Water (70° F.) in 10-foot Gauge Pipe			Hot Salt Water in Boiler. Distilled Hot Water in 10-foot Gauge Pipe		
Boiler Pressure	Temperature in Boiler	Water Level in Boiler *above* Gauge Reading H, fig. 31	Saltness of Boiler Water		Water Level in Boiler *below* Gauge Reading
Lbs.	° F.	Inches	Oz.	$\frac{1}{34}$	Inches
60	307	11	10	1·7	$4\frac{1}{2}$
100	338	$13\frac{1}{2}$	20	3·3	9
150	366	16	30	5·0	14
200	387	18	40	6·7	19

The first of these tables has been calculated on the assumption that the rate of expansion of water is independent of the pressure; but even if not strictly true, both tables show that a serious difference may exist between the gauge-glass indication and the water level, unless

the cold or distilled water in the connecting pipe has first been blown out. Few gauges are so fitted that this can be readily done (see fig. 31), for if the water is low, the opening of the cock D, while B is closed, will not necessarily clear the pipe C D of its water, or at any rate not as quickly as if the cock A had been closed. This being inconvenient, some gauge stands are fitted with a large cock at F.

No double bends, capable of collecting water, should be allowed to exist in the gauge steam pipe, as they would cause the glass to show a wrong level.

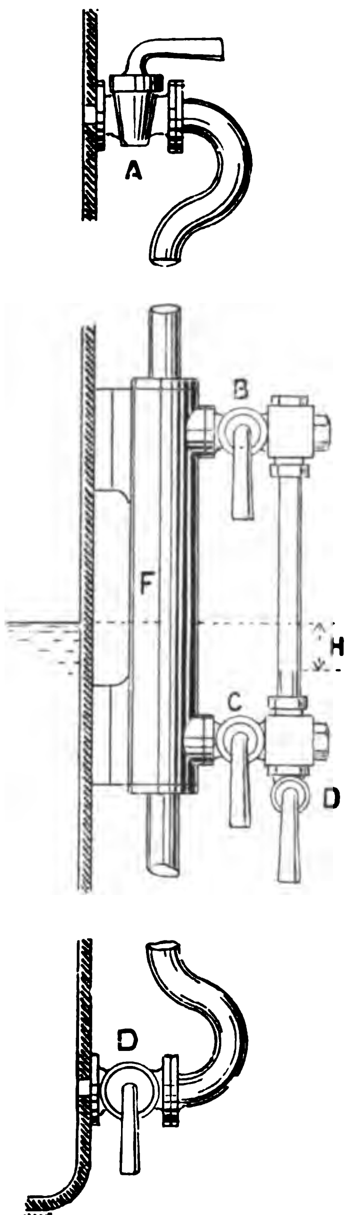


Fig. 31.

Test Cocks are also intended to indicate the boiler water level. They are often fitted to the gauge-glass stand, but their proper position is on the boiler plating. One should be at or near the ordinary water level, a second a few inches higher, and a third on a level with the highest heating surface; for, should the water have been lost out of the gauge glass, it is very important to know whether a mishap can still be averted by putting on all available feed, or whether it is necessary to draw the fires. It is not always well to check the fires, for the immediate effect of stopping ebullition is to lower the water level. Sudden variations of water level are sometimes caused by the drain pipe from the superheater having been fitted below the water level of the boiler and not being closed on occasions when this was necessary; and more recently similar effects have been produced by negligence in not closing the feed-pipe suctions from the boiler bottoms, which are only intended to be open while in use, when steam is being raised.

In marine boilers shortness of water is not necessarily as dangerous as in land boilers, because the furnace crowns are many feet below the ordinary water level, and the upper heating surfaces are not exposed to a very high temperature; besides, in rough weather they are being constantly moistened by water splashing on them. But the case is different in

calm weather, if the steamer has a strong list; for then, even if there is a reasonable height of water in the glass, one of the wing combustion chamber tops may be laid dry for a long period and get overheated. Furnace crown plates would grow red-hot in about five minutes after being laid bare, while combustion chamber top plates would require twice or four times as much time. Obviously this danger is intensified if the gauge glass is fitted to the boiler side, and even the trim of a vessel may expose parts of the heating surface to this influence if the boiler is a very long one. In such cases it is advisable to fit fluid levels in the engine room, which will indicate how much water ought to be in the glass. They are very simple instruments, and can easily be made with the help of a few glass tubes. (See figs. 32, 33.)

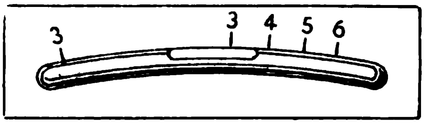


FIG. 32.

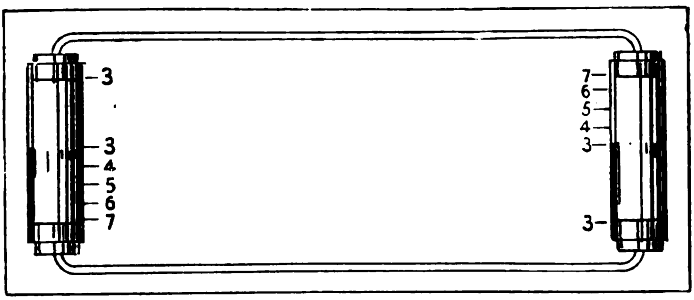


FIG. 33.

Fusible Plugs.—To guard against the troubles arising from shortness of water, some engineers fit fusible plugs to the combustion chamber tops, and in some countries this practice is compulsory; but there is this serious objection, that when a plug is once fused the boiler cannot be used again for some hours. On a ship with only one boiler such a mishap might lead to her total loss, and almost under any circumstances which are likely to occur at sea, it would be better to have the use of boilers which can be worked even if only at a very low pressure, than not to be able to use them at all till a new plug has been inserted. (See p. 18.)

Safety Valves.—Another very important boiler mounting is the safety valve. Its action and design are so simple that little need be said about it, but being practically always out of use, there is a great danger that it will be found out of working order when it should act, and to ensure that this shall not happen, all the working parts should be lined with brass, and should be loose fits, and their working condition should often be tested by lifting the valves from their seats or turning them round. As it has sometimes happened that the valve seats have lifted with the valves, thus preventing any escape of steam, they should always be securely bolted or pinned down.

As long as low pressures were customary, weighted safety valves were common and worked satisfactorily; but now, since high pressures

are almost universal, dead weights have been replaced by springs, for otherwise the working pressure would have to be reduced about 30 % in rough weather, on account of weights being thrown up under the combined action of the steam and the pitching of the vessel. (Wilson, 'Marine E.,' 1892.)

Formerly spring loaded valves were very inefficient, allowing the steam pressure to accumulate considerably, unless the springs were made exceptionally long. This difficulty has been more than overcome by utilising the reaction of the escaping steam, and, instead of allowing the pressure to rise after steam has commenced to blow off, there are many valves which will not close until the pressure has been seriously reduced.

The principles on which such safety valves can be constructed are shown in figs. 34, 35, 36. In the first of these the valve diameter is d,

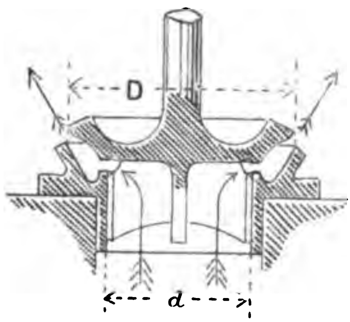


FIG. 34.

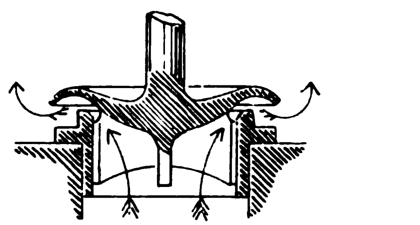


FIG. 35.

but when the steam is escaping it acts on the diameter D, and keeps the valve well open until the pressure has dropped considerably.

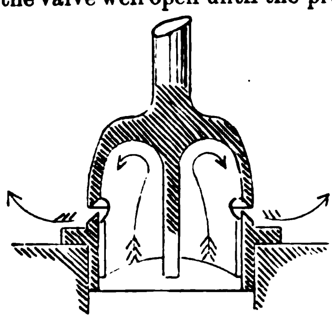


FIG. 36.

In fig. 35 the escaping steam reacts on the surrounding lip, while in fig. 36 it reacts, as shown, by striking against the valve top.

The object of these various designs is that the valve should open fully when it does so at all, that it should not close till the pressure has dropped a pound or two below the working pressure, and that this should be effected with as short a spring as possible.

Obviously all these points are affected by the angle of the seat and the curvature and size of the lip; but, besides that, the diameter and length of the waste steam pipe are influential factors, and it often happens that safety valves which worked satisfactorily when tested at the works, will not do so when fitted on board. The chattering noise which they make is evidently due to the length of the pipe as well as to the weight of the valve and the elasticity of the spring. Every time that the valve comes down and stops the flow of steam, a partial vacuum is formed by the uprushing column, and the valve is lifted up again. The possible remedies are, to have an opening at the lower end of the waste pipe, through which air will

flow whenever a vacuum is formed, or to fit frictional appliances in the lower part of the pipe, such as a series of diaphragms, wire brushes, or a box of pebbles.

Funnel Vibrations.—A related phenomenon is the buzzing noise made by boilers when all the tubes are clean and the fires are alight. The action is evidently the same as that which takes place in an organ tube, only the periods of vibration are slower; but they evidently depend on the height of the funnel.

Alternate Heating and Cooling of Plates.—Very exhaustive and interesting experiments on this subject were made by E. Wehrenfennig ('Organ,' 1884, vol. xxi. p. 216, &c.) He found that heating various metals and cooling them had certain effects, which were reproduced on repeating the experiment, and which were intensified the longer the heating lasted and the higher the temperatures. His limits were boiling water and red heat.

His results are that steel and iron bars and plates shorten, but grow thicker if heated and cooled, about $\frac{1}{20}$ to $\frac{1}{10}$ % for one red heat.

Cast-iron and copper bars and plates lengthen, but grow thinner if heated, say, $\frac{1}{20}$ % and $\frac{1}{5}$ % respectively. Col. H. Clerk ('Proc.,' 1863, vol. xii. p. 452) and H. Caron ('Comp. Rend.,' 1863, vol. lvi. p. 828) arrive at conclusions opposed to the above, but in their cases the cooling was effected suddenly.

E. Wehrenfennig also refers to experiences in connection with locomotives—for instance, that certain fittings grow tighter after a short use if exposed to heat; that iron nuts cannot be unscrewed from copper stays after heating, but brass nuts can; that blisters crack on the fire side, being exposed to greater changes of temperature; that rivet holes of seams in iron or steel furnaces crack, in copper plates they do not do so. Possibly his most interesting fact is that a land boiler contracted so much in length that this led to various troubles about the fittings.

The same thing evidently happens with the through combustion chambers of double-ended boilers, and cases have repeatedly occurred where, on removing the rivets in the circumferential furnace seams, the holes were found to be somewhat blind, although drilled in place, and then, having been rimered fair and re-riveted, they soon grew as blind as before.

Even flanged furnace saddle seams show the same wandering of rivet holes, but, for fear that this statement might be used to shield bad workmanship, it is as well to point out that in all such cases there is a regularity in the blindness of the holes which does not exist if the holes were drilled carelessly out of place.

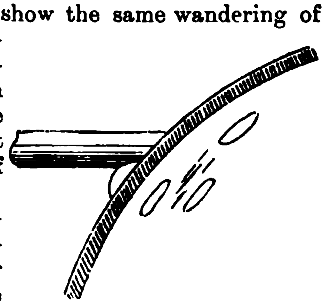


FIG 37.

It has been suggested that the cracking of rivet holes is caused by the carbonising of the iron and steel, due to their contact with fuel or hydrocarbons of the gases; but as an addition of carbon increases the volume of the metal, it is difficult to see how this can set up any tension stresses. The four accompanying sketches show cracks as they occur under palm stays (fig. 37),

in the flat parts of combustion chamber backs when covered with scale (fig. 39), in combustion chamber back plate seams (fig. 38), and in furnace saddle seams (fig. 40).

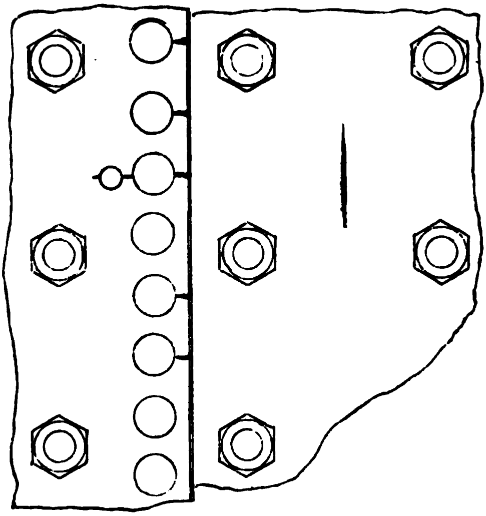


FIG. 38. FIG. 39.

Deterioration of Materials.—That this cracking of rivet holes is due more to the heat than to the impinging action of the flame, is shown by the fact that the joint of the tube plate with the furnace saddle does not give trouble if kept out of the fire, even though, as in fig. 41, the landing is exposed to the impact of the flame, while joints

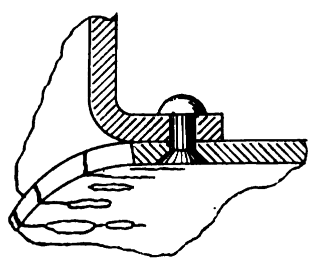


FIG. 40.

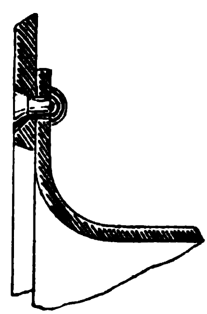


FIG. 41.

which are removed from this action by the depth of a furnace corrugation (fig. 42) suffer. On the other hand, it is unreasonable to imagine that the carbon and other impurities of the fuels will not enter the boiler plates if acted upon sufficiently long (see p. 113); at any rate Lowmoor and other good quality irons do not flange well, and are generally not up to their standard if taken out of the furnaces of old boilers. Fatigue and other influences to which boiler plates are exposed may

make them brittle, but all these points are fully discussed in the chapter on 'Strength of Materials.' Here it will only be necessary to mention that the cracks which are sometimes found at the back-end side flanges of furnaces (figs. 43 and 44), in double-ended boilers, and in the flanges of Adamson's rings (fig. 45), are believed to be due to fatigue – i.e. alternate compression and tension stresses; but other causes, such as inefficient annealing or other bad workmanship, may first have made the plates brittle.

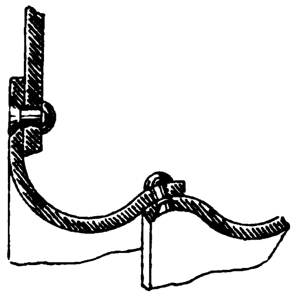


FIG. 42.

As regards the **influence of time**, little information is as yet available, but that it produces effects is strongly supported by the experiences published by A. J. Maginnis in 'Engr.,' 1885, vol. lx. pp. 447, 475, 504, to the effect that the combustion chamber plates of his boilers cracked after being in use for several

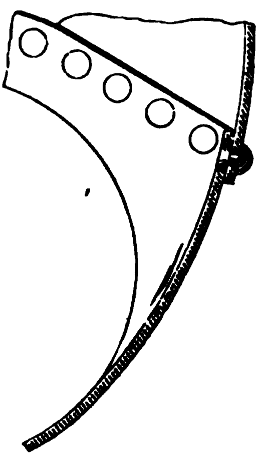


FIG. 43.

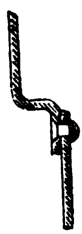


FIG. 44.

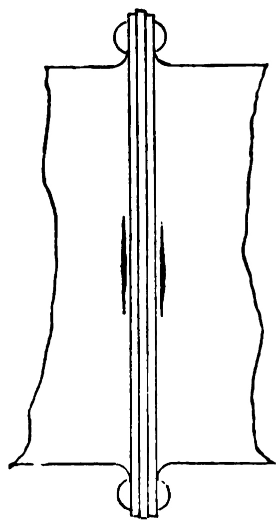


FIG. 45.

years, and that the shells, neck-pieces, and other parts were thoroughly brittle when breaking up the boilers. Here neither fatigue nor gases can have been active. In order to guard against these various troubles care should be taken that all material used for boilers and their repairs should be good—i.e. capable not only of standing certain prescribed tests, but also of resisting the various influences mentioned above—and the general opinion is that, except as regards tenacity, the very mildest qualities are by far the best.

Seams in Furnaces.—Riveted joints should not be exposed to the direct action of the flame. Where this cannot be helped, as in the case of patches, the greatest care should be taken to keep the

seams cool. Scale ought not to be allowed to accumulate there, and the two plates should, if possible, be brought into metallic contact, by removing the black scale, by thoroughly washing away the oil used for drilling (for this is one of the worst conductors of heat), and by fitting and bolting the plates so firmly together during riveting that caulking is almost superfluous. It is perhaps due to the more perfect contact between the plates that double-riveted seams behave better than single-riveted ones when exposed to the direct action of the flame.

Experience has shown that for furnace saddle seams all these precautions are unavailing under the action of forced draught, and it has been suggested that when they must be repaired in this way, instead of using countersunk rivets, very large snap-headed ones should be fitted (fig. 46), with the object of reducing the exposed surface of the outer

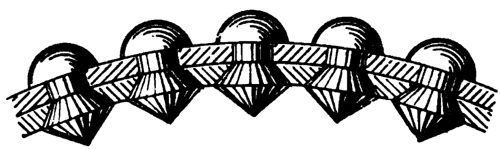


FIG. 46.

landing as much as possible, and at the same time carrying off all the heat through the rivets into the water. (See p. 98.)

Whenever possible, patches should be fitted on the fire sides of defective plates, so that when the rivet holes crack, as shown in fig. 47, it

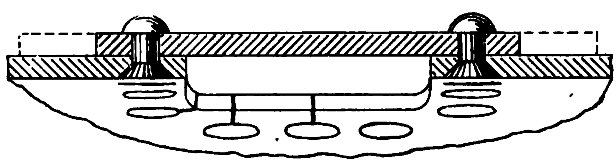


FIG. 47.

will not be necessary to cut out more material and fit a larger patch, as shown in dotted lines, when a renewal is necessary.

Boiler Scale.—One of the chief causes of these rivet holes cracking is scale, and whenever possible it should be removed from these patches. In the chapter on 'Heat Transmission' explanations will be found why the general efficiency of a boiler is not seriously affected by accumulations of scale, but that this substance will cause a considerable rise in the temperature of the plates it covers, which leads, as has just been explained, to cracking or to other troubles, such as leaky tubes and collapsed flues. The idea that scale prevents corrosion may still prevail, and may even be a true one; but the more rational view, that there are other and better means of stopping this waste, is gaining ground, and evaporators, for the supply of distilled water, are coming more and more into use. The removal of one evil often produces another, and recently much damage has been done when using fresh water by oil deposits, which, like all scales and other non-conductors, effect a considerable rise in the temperature of the plates they cover. This neces-

sitates the addition of oil filters, which are inserted between the pumps and the boilers. A description of one of these, together with numerous facts and analyses of boiler deposits, will be found in a paper read by Mr. Edminston, 'N. E. C. I.,' 1892, vol. viii.

Collapsed Furnaces.—One of the greatest mishaps to a boiler which can occur at sea is undoubtedly the collapsing of its furnaces. Generally, but not always, this is due to ignorance or carelessness of the engineer in charge. For instance, it may happen that things have been left in the boiler which, by means of the water circulation, are landed on some plate exposed to the direct action of the flame, and may cause a local bulge, a ludicrous instance being where the head of a dead horse had been put into a Cornish boiler to prevent corrosion, and had settled on the furnace crown. With steamers it has happened that while ashore, the supplementary feed which had to be put on, carried with it mud, sand, or infusorial earth, and this settled on the furnace saddles, or near the upper part of the flanges of Adamson's rings. Under specially favourable conditions the bulges which have then been produced were shaped similarly to the section shown in fig. 48.

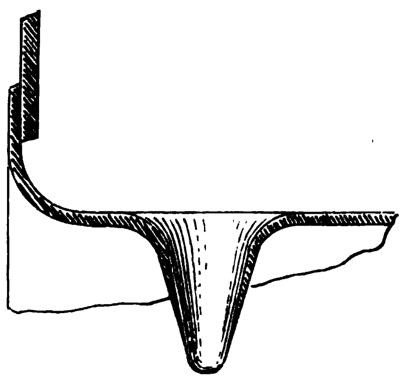


FIG. 48.

A very serious case of collapsed furnaces occurred in a ship with four elliptical boilers. One of them had given trouble, had just been repaired at sea, and when put into use again, all the furnaces in the three other boilers came down together. No doubt the indicated pressure of the repaired boiler was fictitious (see p. 2), and due to air in the steam space, although the water was still fairly cold. On connecting the boilers it disappeared, and was perhaps reduced to a partial vacuum, into which steam suddenly rushed from the other boilers. The consequent excessive ebullition, aggravated as it was in these boilers by very restricted water spaces, and very little clearance below the tubes, must have raised the water away from the furnace crowns, and these collapsed.

Very mysterious collapses are sometimes caused by emptying and refilling a boiler quickly in port, without removing the manhole doors or cleaning its inside. It would appear as if, under these conditions, the scale on the tubes falls off, lodges on the furnace crowns, and, as it is not removed, causes overheating. It also happens that on blowing out the boiler, the scum and oily matter which floated at the water level adhere to the heating surfaces, even when the new water is admitted.

By far the greatest number of collapses are due to scale, salt, or greasy matter; but then, instead of the furnace crowns coming down, it is the sides which come in; for, although the deposit may be uniformly distributed, the heat of the fire is greatest on either side just over the burning fuel. Shortness of water and local deposits bring down the crowns, not the sides. A mystery about all these accidents

is, that generally several furnaces come down at about the same time, even though they are fitted in different boilers.

Bulging of Flat Plates. - Another part of the boiler where scale deposits produce visible deformations is the flat plates of the combustion chambers. With narrow water spaces the presence of much steam assists in causing the plates to get hot and bulge, by keeping the scale partially dry. If this treatment did not tend to make the plates brittle, and cause them to crack between the stays, little harm would be done, because in their bulged shape plates are stronger than when flat. Possibly, too, the stretching of the plates enlarges the stay holes, and causes these to leak. When the bulging is serious the favourable conditions for further deformation are increased, because it is now more difficult than before to remove the scale.

In boilers with two or four furnaces, leading into one large combustion chamber, serious bulging sometimes takes place at the top of the saddle between the two central furnaces, but only if they are exposed to the heat of the fire, and if several angle irons have been fitted underneath, as is customary for strengthening these parts ; the enclosed spaces form steam pockets, and the saddle plate above and between them gets overheated.

Tube-Plate Troubles in modern warships have recently attracted much attention, but it does not appear that any satisfactory explanations have been put forward. Doubtless here, too, it is a case of overheating caused by a high temperature in the combustion chamber, and an excess of steam bubbles near the plate on the water side. The phenomenon (see p. 23) that iron contracts permanently when heated and cooled again might account for the tubes shrinking and the tube plate drawing away from them, but it is difficult to suggest a remedy which has not been tried and failed.

Experiments on the subject have been made by A. F. Yarrow ('N. A.,' 1881, vol. xxxii. p. 106), who showed that tube plates tend to bulge towards the fire side, and thus to draw away from the tubes ; but the strong curvatures which he was able to produce are not met with in practice. An idea of the temperature of a tube plate will be obtained by examining the following case. (Compare p. 94.) Let the thickness of a tube plate (fig. 49) be 1 in., if perfectly clean its temperature would be 40° F. higher on the fire side than near the water, while transmitting sufficient heat to evaparate 20 lbs. of water per square foot per hour. The tube, being only $\frac{1}{16}$ in. thick, will be 4° F. hotter on the one side than on the other, and therefore its average temperature will only be 18° F. less than that of the tube plate, which corresponds to a relative shrinkage of about $\frac{1}{3000}$ in. This can be neglected in comparison with the compression existing in the tube, due to the expanding. If covered with scale, as shown in fig. 49, or if in contact with a film of steam, so that the water can only reach the plate with difficulty, then the conditions are changed.

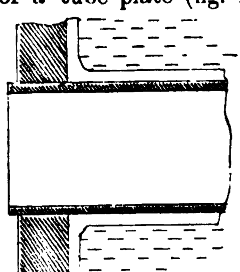


Fig. 49.

Suppose that the scale is $\frac{1}{16}$ in. thick, and that only half of the above-mentioned quantity of heat is being transmitted, then the metal

of the tube and tube plate would be about 1,000° F. hotter than the boiler water. But it is difficult to believe that the heat transmitted through the tubes is equal to that passing through the plate, and certainly, within a very few inches from the end, the difference will exceed 50 %, so that it is not unreasonable to assume that the tubes are only 500° F. hotter than the water. With 3-in. diameters this corresponds to a relative difference of $\frac{1}{100}$ in. between the tube and hole. The heightened temperature will reduce the elastic limit to, say, 13 tons, so that the pressure, due to the expanding, cannot dilate them more than $\frac{3}{1000}$ in., thus leaving an opening of half the difference, viz. $\frac{7}{2000}$ in., or about $\frac{1}{300}$ in. all round the tube. This is sufficient to cause serious leakage. Salt from the boiler water is supposed to choke this opening as long as the heat is fierce, but dissolves out and renews the leakage when cold. (A. J. Durston, 'N. A.,' 1893, vol. xxxiv.)

In most boilers where this has occurred another force was at work intensifying the leakage. It is well known that in order to reduce the crushing stress on tube plates of double-ended boilers, not only the sides, but also the tops and bottoms of the combustion chambers, are stayed to the shell (fig. 50). When pressure is raised in the boiler its diameter increases, and tension stresses are produced in the tube plates, which they, with their small effective sections (only about 25 % of the solid plate), are unable to resist, except by relieving the tubes of a little of the pressure which holds them in place. At the same time the tubes contract their diameters a little, due to the external steam pressure.

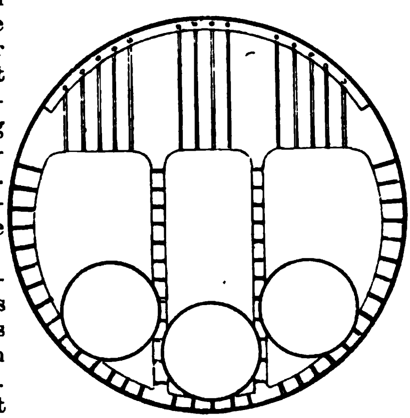


FIG. 50.

The chief difficulty in accepting these views is that the leakages are not diminished when the fires are drawn, and the phenomenon noticed by Wehrenfennig (see p. 23), that iron and steel contract permanently when heated, even if only up to boiling point, may help to explain the matter. The author's experiments on iron and steel bars which had been drawn out cold under a hammer, on cold rolled bars, on wire (drawn), and on stretched test pieces, show that they all contract permanently on being heated. The distortion which takes place during the annealing of flanged furnace front plates, points to the same conclusion, viz. that leaky tubes are caused by a partial annealing of the expanded tube metal, due to excessive heat. The remedy which at once suggested itself is to anneal and re-expand the tube ends in place. According to this view, tubes which have leaked once (and have therefore been annealed) ought not to leak again after re-expanding, but although they do, the trouble is lessened. It is curious that tubes which have leaked on account of excessive scale grow tight when it has been removed, even if the ends are not re-expanded.

The only effective remedy for these troubles, if they were not caused

by scale, appears to be the use of copper tube plates. Possibly a wider pitching of the tubes, or reducing their diameters at the back end, may do good. Or the tube ends might be subjected to a preliminary compression. This, however, would require an amount of care in boring the tube plate holes which is not often bestowed on them. Differences of $\frac{1}{16}$ in. in the diameters of various holes in one plate are not uncommon, and most of them are distinctly oval, sometimes as much as $\frac{1}{32}$ in. Taking more care to make the holes circular and of equal size may reduce the trouble.

In a recent experiment (see discussion on A. J. Durston's paper, 'N. A.,' 1893, vol. xxxiv.) a compressed tube was compared with another which had been expanded in the usual way. Both were fixed in a tube plate which was placed over a smith's fire and heated to a temperature at which about 100 lbs. of water were evaporated per square foot per hour, but the plate was only occasionally covered with water, which then remained in a spheroidal condition. After a few hours' exposure the plate was allowed to grow dull red hot, when both tubes grew slack. Careful measurements showed that the expanded tube had contracted its diameter $\frac{1}{40}$ in., while the compressed one had not altered, but its hole in the tube plate had changed its taper from 1 : 12 to 1 : 10. This compressed tube could not be withdrawn out of the hole, for its projecting end ($\frac{1}{8}$ in.) had expanded during annealing. This goes far to prove that leaky tube ends are caused by raising their temperature to redness.

Attempts are being made to electrically weld tubes into the plates, but the risk of burning an internal part of a boiler which is nearly finished must deter all except the most venturesome from trying this plan.

As a makeshift ferrules are driven into the ends of boiler tubes (figs. 51 and 52), but they soon burn away. Better results are said to have been obtained by making the ferrules of cast instead of wrought iron, and leaving an air space between them and the tube (see fig. 53 and A. J. Durston's paper).

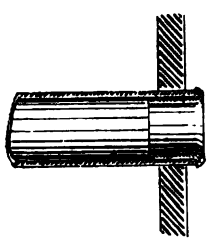


FIG. 51.

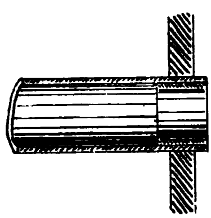


FIG. 52.

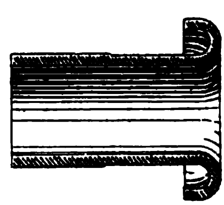


FIG. 53.

Cleaning Boilers.—From the foregoing remarks it is clear that many troubles are caused by the accumulation of scale, grease, or even salt, and where this cannot be prevented, great care has to be taken that the boiler shall be cleaned out and scaled as often as possible. This work is never done well unless the various parts of the boiler are easily accessible, not only to boys, but to the engineer, who has to see that the work is properly done. None of the water spaces between the tubes

should, therefore, be less than 10 ins., and if that amount of space cannot be provided over the wing furnaces, it ought to exist between the tubes and the shell, or a manhole should be fitted in the two wings of each boiler. They should not be made smaller than $10\frac{1}{2} \times 14$ ins., but 12×16 ins. is ample. A sufficiently large manhole should also be fitted between and under the furnaces, either at the front or back end of the boiler.

Corrosion.—This subject is fully dealt with in a separate chapter, but here it will be necessary to make a few remarks on the external wasting away of plates. This never occurs except in the presence of moisture, and is found chiefly near manholes, near leaky joints, at the boiler bottoms and ends, where they come in contact with bilge water, and above all near the furnace fronts and combustion chamber bottoms. These are exposed to the very injurious action of moist ashes, containing large percentages of sulphuric and other acids, which are partly liberated by the ash-cock water. By keeping the boilers perfectly tight, and protecting the front plate from moist ashes, all these troubles can be avoided. Naturally also leaky tubes and seams cause external corrosion, for then a combination exists there of heat, moisture, salt, and noxious acids.

As regards manholes, an excellent practice is gaining ground of flanging them and facing the edges and grooving the door, as shown in fig. 54, which permits of their being faced up again when worn out.

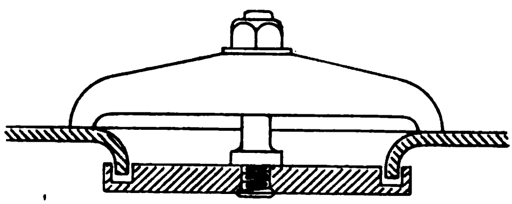


FIG. 54.

The leakage from valve and other flanges, the drainage from test cocks, and the moisture which collects in the wood of platforms adjoining the boiler backs, all tend to shorten the life of a boiler, and should be prevented as much as possible.

Boiler Repairs.—Sooner or later even the best managed boilers will need repairs. If badly done they lead to further troubles, and may prematurely necessitate the renewal of the entire boiler. Care should therefore be taken that the workmanship is of the best, and that no structural defects are introduced, which would either prevent circulation or hinder the removal of scale.

Bulged Furnaces.—The most serious troubles with new boilers are the furnaces. A careless engineer, or other causes, which have been already mentioned, may suddenly bring them down. If the deformation is not great, i.e. only a few inches, and spread over a considerable area, the best remedy is to press the plates out again. Attempts have repeatedly been made to do this cold, but the deformation can never be perfectly removed in this way, and soon returns. Besides, other parts of the furnaces are thereby strained, and it has happened that while

pressing up a furnace crown the welded seam at the bottom cracked and had to be patched. Good results are obtained by heating the furnaces locally, and then pressing them out, but there is the danger of making the plates brittle by this treatment. As a safeguard such furnaces should be heavily hammered twenty-four hours after they have cooled ; if very brittle they will crack. In order to guard against such injurious effects it is well to reheat the plates after they have been pressed out hot, for local heating seems to be injurious only when accompanied by straining of those parts. (See p. 203.)

Satisfactory results have been obtained by using a cast-iron mould for a head to the press. It must be made of a substantial thickness, heated to redness and then applied, and left in position till cold. The correct shape of the furnace is very soon regained, and while slowly cooling the hot cast iron anneals the plate. Only screw jacks should be used for this work, because the hydraulic ones can never be kept perfectly tight and have to be pumped up occasionally, whereby strains are produced during the period of cooling, which it is the object of this proceeding to avoid. Furnaces which have once come down very often do so again unless they are strengthened by rings.

Collapsed Furnaces.—If the furnaces have collapsed thoroughly they must be renewed, and it depends very much upon the boiler design whether this can be done efficiently. If the furnace back end was flanged inside of the tube plate, as is now generally the case, there is hardly any other remedy than to cut the back end and to insert an unflanged furnace (figs. 55, 58). Sufficient width should be left for a

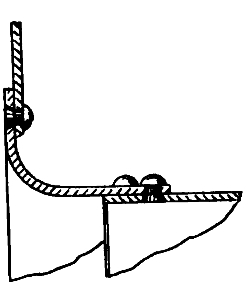


FIG. 55.

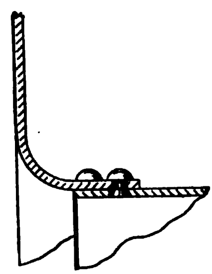


FIG. 56.

double row of rivets, for such a joint seems to give better results than a single-riveted one. Chiefly for cheapness' sake, but partly for facilitating the above-mentioned repairs, the tube plates, instead of the furnaces, are sometimes flanged (fig. 56). Should the furnace collapse, it is then only necessary to draw it and to fit a new one. Under any circumstances no expense or trouble should be spared on these seams to bring the two plates into metallic contact. It is well to wash the two surfaces in sal ammoniac, so as to remove the scale; also, when all the holes have been drilled, which should be done in place, the furnace ought to be partly withdrawn, the burrs removed, as well as every trace of oil, because this is a very bad conductor of heat. The riveting should not be started from the bottom of the seam ; otherwise, before the top is reached the plates at the crown will be far apart (see fig. 57). It is

then practically impossible to make the seam permanently tight, and after being in use a short time it is sure to leak again. Sometimes it only opens on the fire side, and then, though the blade of a pocket knife can be inserted up to the other caulked edge, no leakage seems to take place. In order to be quite safe, it is best to fit and rivet the furnace crown into position before the furnace bottom is introduced. The longitudinal seam should be slightly inclined, as shown in fig. 58, and then it is an easy matter to make it

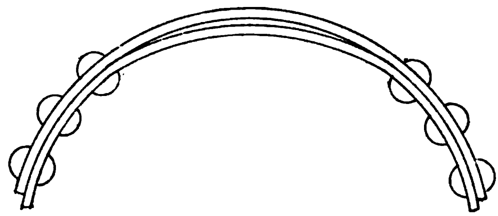


FIG. 57.

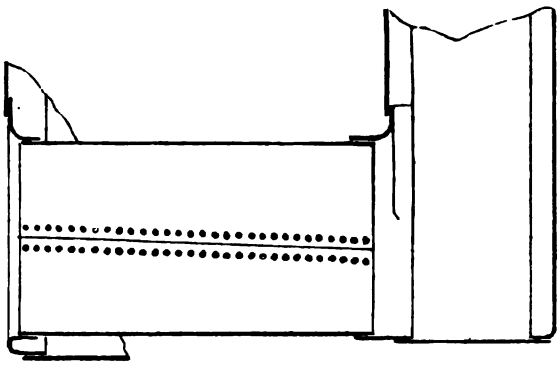


FIG. 58.

a very good fit. Another way of repairing a collapsed furnace is shown in fig. 59. The furnace crown is cut in two, and the forward end is

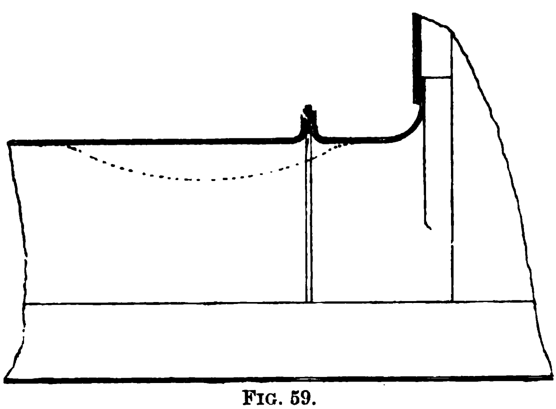


FIG. 59.

renewed, while the saddle is taken out, flanged, as shown, and refitted with an Adamson's ring.

With a furnace whose back end is not flanged, repairs may sometimes be effected by removing all the rivets and turning it round, so that the saddle seam is placed in the ashpit. The weld which was at the bottom of the furnace is now somewhere near the crown. Central furnaces can, under certain conditions of shape, be renewed by first removing the combustion chamber bottom plate.

If the furnace saddle is flanged over the tube plate, and if the furnace front plate is outside the front tube plate (see fig. 60), then

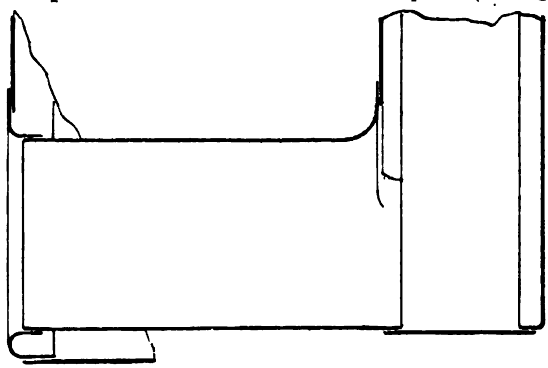


Fig. 60.

both it and the furnace can be removed, and the one renewed. But if the furnace front plate is placed inside the front tube plate, and has to be withdrawn, it is necessary to cut away its two corners (fig. 61), so as to be able to tilt it out of and into position. The amount of metal to be removed depends on the depth of the flange. This plan cannot be carried out on boilers with three furnaces, unless the horizontal seam intersects the central bundle of tubes.

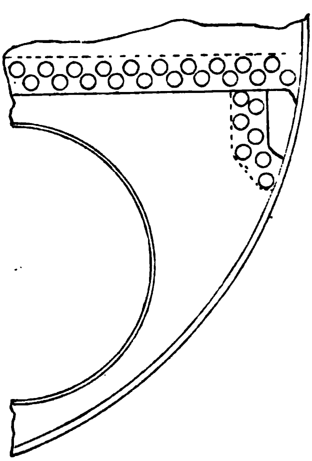


Fig. 61.

The arrangement shown in fig. 60, which facilitates these repairs, has the further advantage that both the back and the front seams are very easily caulked from the outside. Its only drawback is that one loses about 1½ in. of tube length.

Flanged furnaces can occasionally be removed and replaced intact by cutting away the lower part of their back ends (fig. 62). The wing combustion chambers would have to be specially constructed, so that no part of the furnace flange is wider than the front ends of the furnaces (fig. 63).

Before attempting such repairs, careful measurements should be taken, and with the help of a board, representing the furnace front, and a wooden model of the furnace (fig. 64), the action of passing the

latter through the former should be performed. This will also enable one to determine how much of the back end has to be cut away.

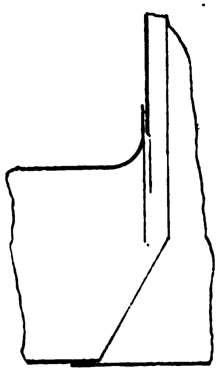


FIG. 62.

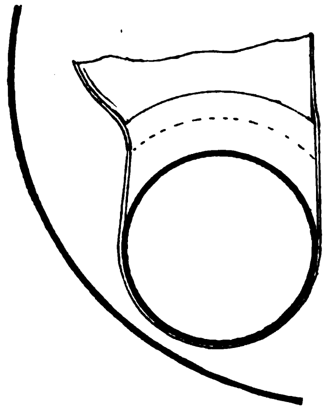


FIG 63.

Furnace Patches.—Furnaces sometimes suffer severely from pitting along the line of fire bars. If some of the pit holes should have pierced the plate, it is necessary to drill them out and fit screwed studs, or, better still, broad-headed rivets, which may be square, oval, or irregular in section, to suit the special case. Neither give trouble by leakage, but they do not add to the strength of the furnace, and, unless absolutely necessary, it is better not to perforate it at any point above the bars. If the pitting is uniformly distributed, as well as deep, there is danger of this part of the furnace giving way, but as yet no such case seems to have happened. To guard against this, or to prevent further pitting, some engineers bolt doubling plates, well bedded in red-lead cement, along the line of fire bars (fig. 65). It is difficult to imagine a more inappropriate remedy, for to place here, at the hottest part of the furnace, not only two thicknesses of plate, but actually to separate them by a highly non-conducting material, is little better than inviting disaster. If any action is necessary, the furnace crowns should be removed.

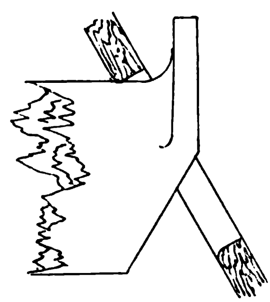


FIG. 64.

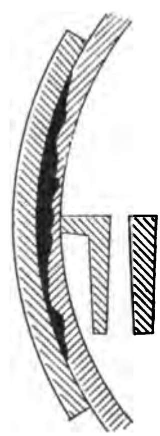


FIG. 65.

Furnace bottoms are sometimes doubled in the same way ; but this is not a permanent repair, and the plates should occasionally be removed for examination (see p. 37).

Blisters in the furnace crowns need only be pared away, unless they are so large and deep that there is fear of their weakening the furnace. In that case they have to be cut out and a patch fitted.

Cracked Furnaces. -Plain furnaces have hardly ever been known to crack except at the flanges or the seams, but every one of the patented forms has done so. If due to scale or grease, the cracks start on the fire side and sometimes run longitudinally; if due to unequal expansion or excessive strains, they sometimes show first on the water side, and generally in a circumferential direction. Similar cracks (see fig. 43, p. 25) are sometimes found in the sides of the furnace saddles, and can often be repaired by chain-pinning them. They make their appearance on the fire side, and sometimes, if not interfered with, do not penetrate through the plate. It is difficult to say whether they are due to excessive strains or to injured material, such as working the flanges at a blue heat, or to both causes. The manufacturers take great pains to thoroughly anneal the patent flues before sending them out, but unfortunately boiler makers find it more convenient to reheat the corners while fitting them to the tube plates than to alter the latter to the required shape. In some works these corners are even welded, and possibly not annealed afterwards. But the greatest danger seems to be incurred if heaters are applied to these points. It has been suggested that the furnaces should be heavily hammered after the hydraulic test.

When the cracks, which show themselves, are so serious that tapping pins into them would not prevent leakage, the affected parts have to be cut out and patches fitted. In fig. 66 the patch is fitted on the fire side, and can easily be renewed if the rivet holes should crack. In fig. 67 it

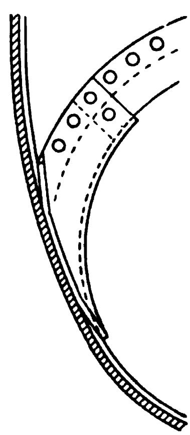


FIG. 66.

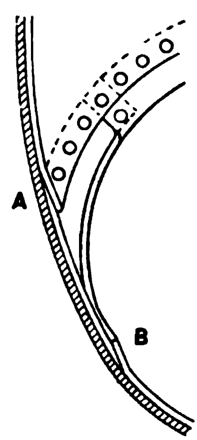


FIG. 67.

is fitted on the water side. This is done with the object of keeping the seam turned away from the fire, but it is a most difficult piece of work, because the seams at A and B have to be sprung open. This patch is also not to be recommended, because if the rivet holes should crack, a larger corner would now have to be cut out and replaced (see fig. 47, p. 26).

Combustion Chamber Bottoms.—These are often very seriously cor-

roded on the fire side, particularly near leaky seams, and, as even the rivet heads retain their shape when wasted (see fig. 68), there is danger that such defects will be overlooked. The salt from leaky tubes when mixed with ashes cause corrosion at these points, and to prevent this, the bottoms are sometimes cemented. This is a very dangerous practice ; not only does this prevent any leakage from being detected, but if there is one, or even if moisture finds its way between the plates and the cement, the furnace heat will convert it into steam and cause a small explosion, which may do damage to bridges and fire doors, &c. Fitting doubling plates to these parts may lead to the same result, but the consequences would be still more serious.

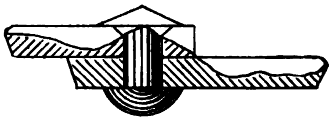


FIG. 68.

The corrosion on the water side of the combustion chamber is often very evenly distributed, and if the bottoms are found to be thin, the backs are most likely worn away too, and before starting to cut out the one it is best to drill holes in the other for measuring the thickness.

The best means for detecting weak places in plates is to make a thorough examination when the plates are well scaled. The shadows produced by the lamp often indicate irregularities, particularly pit holes. The hammer will give no reliable indication of thickness when this exceeds $\frac{1}{8}$ in., and then it is of course possible to dent the plate and even to knock a hole into it. When general wasting and consequently structural weakness is suspected, drilling the plates is the only means of measuring their thickness, though magnetic and electric tests have been suggested and might be made available.

Before cutting out thin plates, it is well to drill test holes beyond the intended new seam, for if further thin places are revealed after the plate has been cut across, it will be necessary to repeat the operation. Patches on the lower parts of the combustion chamber usually include flanges, and care should be taken that these are properly fitted, for none of the seams or rivets can be caulked from the water side, as is the case with new boilers. Of course new stays will have to be fitted, and these should always be nutted, no matter whether this was originally the case or not.

Screw Stays very often lose their nuts by burning, or rather bad welds open under the influence of the heat (fig. 69). It is quite a common practice to cut off the greater part of the projecting stay and rivet it over. This should not be done, because it is due to the presence of the nuts that a reduction is allowed in the thickness of the plates. The better plan is to cut the thread deeper (fig. 70), for which special tools exist, and to fit new nuts.

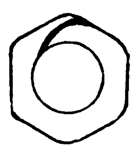


FIG. 69.

This burning away of the stay nuts is generally due to their not being in metallic contact either with the stay or the plate ; or, in other words, they are a loose fit on the thread, and are resting on a washer and two thicknesses of red-lead cement. When refitting the nuts, these mistakes should not be repeated, but where the stays are not normal to the inside plates, taper washers are a necessary evil. This arrangement should be objected to in new boilers.

With thin plates the stay ends often leak, particularly if their ends are riveted over. Instead of renewing them, some boiler-makers

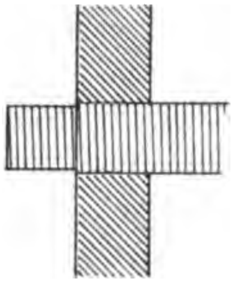


FIG. 70.

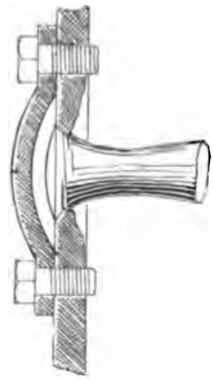


FIG. 71.

go to the trouble of making a small cap and bolting it over the head, as shown in fig. 71. Sometimes several stays in one combustion chamber are hidden away in this manner, and it is only a wonder that this practice has never led to a disaster.

The excuse made for fitting these patches is that the plates had become too thin for holding a new stay. If the thinning is only local, a very efficient repair is to replace the stay by one of a larger diameter. But in screwing up the nut its power will perhaps be so great that the thread in the plate gets stripped; to guard against this, a check nut should be fitted on the inside. It is bad for the transmission of heat, but is better than no support at all.

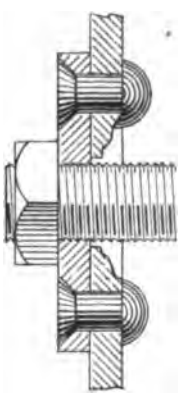


FIG. 27.

Often it will be necessary to remove the stay, and to fit a small patch and tap it, as shown in fig. 72. This is the best possible arrangement, particularly if the stay is not normal to the plate, because the patch could be made taper, bent or recessed, so as to let the nut come into metallic contact with it.

Sometimes the plates remain uninjured while the screw stays waste away and will have to be renewed, generally of a larger diameter, the threads having been injured.

In order to effect these various repairs, and to stop external leakages, sufficient space should be left at the backs of boilers, so that a man could work there.

It will be noticed that those stays which are nearest the combustion chamber bottoms and sides give the most trouble. This is no doubt due to the very severe strains to which they are subjected through the unequal expansion of furnace and shell plate, and therefore they ought to be kept as far away from the flanges as the strength of the boiler back plate will allow (see p. 140).

Similar remarks apply to the top row of screw stays to boiler shells;

if placed too near the combustion chamber top they also are subjected to very severe strains, sometimes leading to rupture.

Tube-Plate Troubles.—The last of the troubles in combustion chambers is the tube plate. An explanation as to the causes of leaky tubes has already been attempted, and some remedies suggested. The repeated re-expanding, together with the injury caused to the metal by getting hot, if covered with scale, ultimately causes the tube plate to crack. Fortunately, since steel and thicker tube plates have been introduced, such cracks are rarely met with. If caulking will not stop the leak, or if several spacings show cracks, it is necessary to bolt little spectacle patches (fig. 73*a*) over them, preferably on both sides. Keys, shaped like a dumb-bell (fig. 73*b*), are said to have given good results, but the fitting must be carefully done to make it a good job.

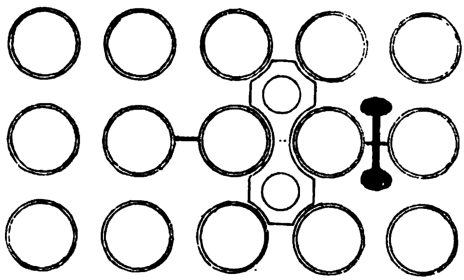


FIG. 73*a*. FIG. 73*b*.

If the front tube plates of boilers are curved or bent back at the top, they are subjected to very serious tension stresses, which in one case led to a rupture, so that it was found necessary to fit vertical stays, as shown in fig. 74. It is, however, questionable whether the stays were of much value, as their lower extremities had only been bolted to the furnace tops, which, being thin, could not have offered a resistance if the large stays had been strained to their utmost.

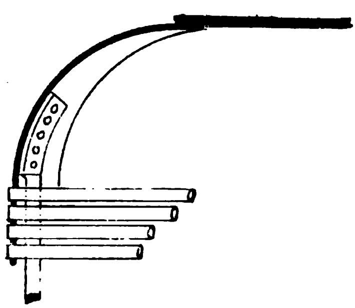


FIG. 74.

Repairs to the Boiler Shells are sometimes necessary on account of the wasting away or cracking of circumferential seams. As such patches are not required for strength, but only for water-tightness, they need not be thick. At one time it was customary to make them of cast brass, a thick sheet of lead, which was hammered over the seam, being used as a pattern ; but it was found that this gave rise to very serious local corrosion, caused, perhaps, by galvanic action. In their stead light iron cover plates, filled with cement, are bolted over these seams, as shown in fig. 75. They were very efficient in stopping leaks where caulking was unavailing. Of course if the shell plate is seriously weakened it is necessary either to fit an efficient doubling plate, or the defective part must be cut out, and a

FIG. 75.

proper patch put in its place; but it is very seldom that this occurs, except perhaps with donkey boilers, whose bottoms cannot be properly protected from moisture.

It is different with the end plates of main boilers; they are exposed at one end to the hot and moist ashes, and at the other to leakages from flanges and manholes, &c. As these plates are originally very thick, and more than sufficiently supported, their wasting away is not so much a danger as a nuisance. Most of the patches to these parts are therefore simply doubling plates over the thinnest places, and it is only when no other remedy is thought effective that the bad parts are cut out.

The following are a few sketches of repairs which may occasionally be necessary. Figs. 76, 77, show a case in which the circumferential

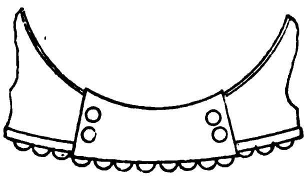


FIG. 76.

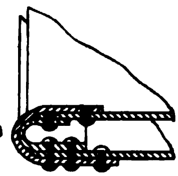


FIG. 77.

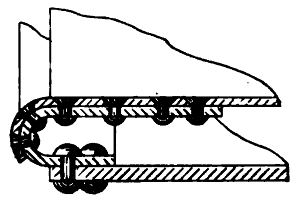


FIG. 78.

seams at the front ends of the shell and the furnace have wasted away and have been repaired by a covering patch. As it forms a ridge on the inside of the furnace, against which the rake will always be knocking, it is better to carry out this repair as shown in figs. 78, 79. Part of the furnace bottom is cut away, and the patch fitted in its place. Of course a strap will have to be placed on the water side, as shown in dotted lines.

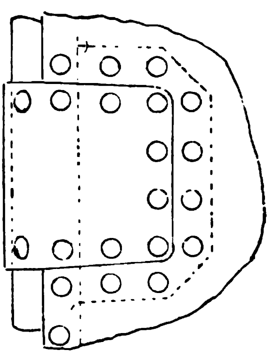


FIG. 79.

If only the front plate is wasted away, and if this is only between the two furnaces, an internal patch can be fitted, as shown in figs. 80, 81. Generally a manhole or a few large stays will be found here, and as it is impossible to get a sufficiently large patch into the boiler to cover them, it will be necessary to fit it from the outside.

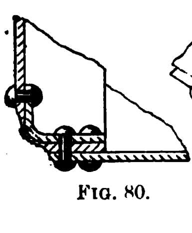


FIG. 80.

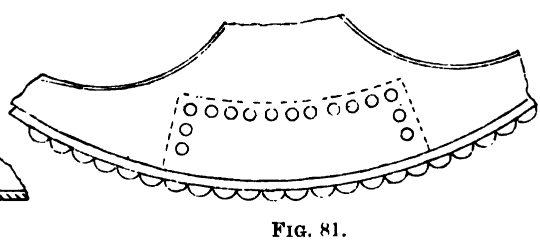


FIG. 81.

When the lower front seams both of the furnaces and the shell are in a bad condition, it may be necessary partly to renew the front plate,

and to make its flanges sufficiently deep to take in. another row of rivets. The joints should be placed where there is sufficient room for working at them. Lap joints, as shown in fig. 83, are very difficult to make, for if looked at from the back (fig. 82), it will be noticed that

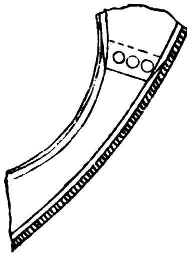


FIG. 82.

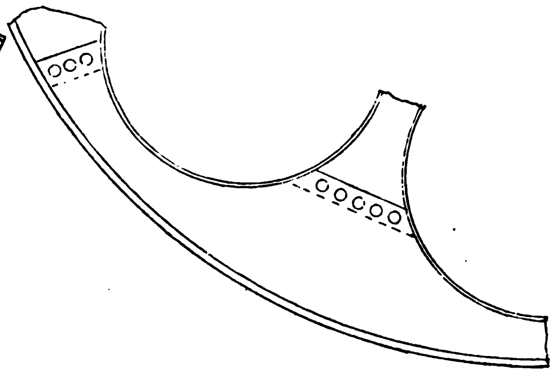


FIG. 83.

the patch has to be fitted over the remaining plate, which cannot be done efficiently.

A simpler plan is to butt the plates at these points, and to fit flanged butt straps internally (fig. 84), or, where the space is too narrow, a solid block (fig. 85). In either case this part has to be carefully fitted after all the other seams have been riveted up. Of course with the solid block it will be necessary to use screwed studs and bolts, except perhaps for those points marked *c*, for which through rivets may be used, but only if the holes come opposite each other. The flanges should not be cut away parallel to the axis of the boiler and furnace, but should slant downwards (see dotted line, fig. 86); this will enable the plate edges to be driven tight together.

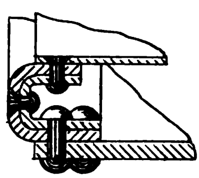


FIG. 84.

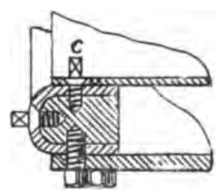


FIG. 85.

Before deciding which plan to adopt, a sketch should be made to see whether it is possible to insert the various rivets. Much space is gained if the rivet holes are countersunk inside, so as to do away with the heads, and outside, so as to be able to use short rivets (see fig. 87).

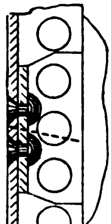


FIG. 86.

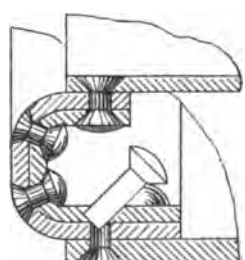


FIG. 87.

Evaporation and Circulation.—Some information on this subject,

particularly on the distribution of evaporation in a boiler and its efficiency, will be found in the chapter on 'Transmission of Heat.' Here attention will be drawn, not so much to the economical side of the question as to its practical bearings on the life of a boiler.

Formerly, when there were almost more boiler explosions on steamers than ashore, while neither iron works nor boiler-makers would admit that it was their fault, speculations about retarded ebullition and similar subjects were freely indulged in. The summary of about twenty papers, dating from 1786 to 1864, is contained in C. Tomlinson's paper ('Phil. Mag.,' 1869). The Franklin Institution made some experiments to show that the sudden opening of the stop valve of an overstrained boiler led to its destruction. A related subject is that known as the action of water hammers, which is met with chiefly in steam pipes, and was evidently the cause of the explosions on the 'Elbe' and at Deptford. A considerable amount of information on this subject will be found in Clark and Colburn's paper, 'Am. M. E.,' vol. iv. p. 404.

Priming.—Information on this subject is as yet almost non-existent. The generally accepted views are that low-pressure boilers prime far more than high-pressure ones, and therefore require more steam space, or perhaps more steam height; also, that throttling the steam at the main stop valve reduces or stops priming, and that the injection of oil, particularly mineral oil, is a still more efficient remedy. Soda and salt seem to increase priming, probably by forming soaps with the oils. The action of mineral oils may be compared to the action of oil on troubled waters: it prevents the formation of light bubbles. That the partial closing of the boiler stop valve has an effect in reducing priming, while linking up the engine, so as to use the same reduced amount of steam, has not, clearly points to the necessity of imparting violent motion to the steam, or rather to the as yet uninjured froth, so as to burst the bubbles.

J. T. Thornycroft ('C. E.,' 1890, vol. xcix. p. 41) argues this point very clearly, and has effectively demonstrated that a very small steam space will suffice, if only the mixture of steam and water is properly guided. In his water-tube boiler he admits this mixture at the upper circumference of the dome (fig. 88), and dashes it against an internal baffle plate, whereupon the water falls to the level W and the steam is carried off through the pipe A. This small dome separated sufficient steam for 774 I.H.P., equal to about 10 cubic ft. of steam per second, while the internal diameter was only 26 ins., and the clear height from water level to crown of baffle plate only 15 ins. An ordinary double-ended marine boiler of the same power would require at least 48 ins. of steam-space height.

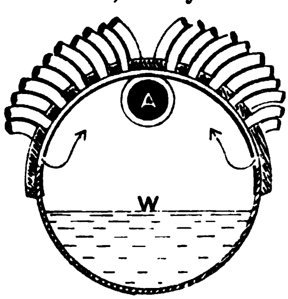


FIG. 88.

That this large space is necessary need not be wondered at, for, in spite of the wide water space between the nests of tubes, which are primarily intended to act as downcast shafts, facilitating circulation, they are very far removed from doing their work properly. Perhaps

more than one-third of the total evaporation takes place at the furnace crowns, but instead of allowing the steam to ascend to the water level as freely as possible, by removing a central row of tubes (fig. 89), it has to rise as best it can (fig. 90), and naturally struggles towards the

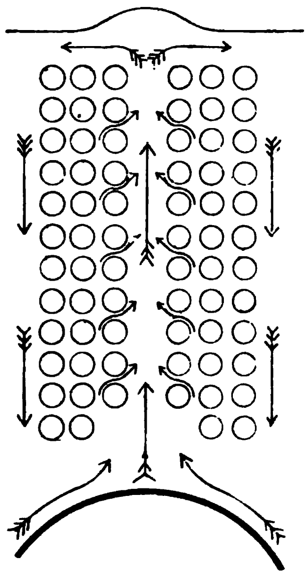


FIG. 89.

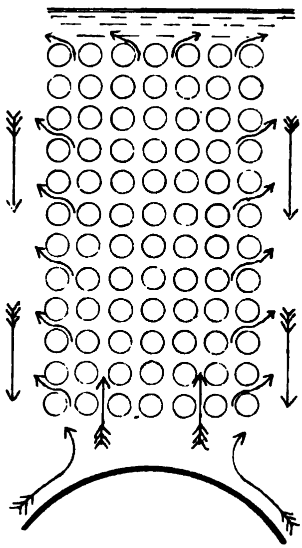


FIG. 90.

water spaces, where it comes in conflict with the downward current. By closing up all but the wing water spaces, this struggle can be made to grow so severe that the imprisoned steam raises the upper water level, as, for instance, in locomotive and Navy boilers. The consequence is that certain parts of the heating surface are oftener exposed to steam than to water, and even though they may not get burnt, are occasionally ineffective.

A half-hearted attempt to direct the current of steam and water is sometimes made by bolting a few plates horizontally to the lower row of steam-space stays; but to be really effective they ought to cover the whole water level, with the exception of the wings, and no downward currents should be allowed at the centre water spaces.

In some boilers flat sheet-iron tubes are fitted to wings, reaching to the boiler bottoms, but it is to be feared that the inducement for the water to circulate through them, while there are wider water spaces on either side, is not a very strong one. Undoubtedly, however, the water which does enter them must fall below the furnaces.

Improvements in circulation, though not a remedy against priming, could sometimes be effected by removing a central row of tubes from each nest, and securing vertical plates to the outside rows of tubes, which would prevent any steam from entering the water spaces. By placing horizontal plates over each nest of tubes the separation of

steam and water would also be effected; but all loose parts in a boiler are a nuisance.

The notions as to what takes place inside a boiler are exceedingly vague, and it may be of interest to draw attention to a few points.

Take, for instance, a boiler 15 ft. diameter and 11 ft. long. Its steam space will be about 4 ft. high and have a capacity of 450 cubic ft. It will be capable of evaporating about 10,000 lbs. of water per hour under natural, and double that quantity under forced, draught. The total area of the water level is about 150 sq. ft., so that under forced-draught conditions every square foot of this water level emits 133 lbs. of steam per hour, or $\frac{1}{28}$ lb. per second. At 180 lbs. working pressure this is equal to a volume of $\frac{1}{12}$ cubic ft., or 144 cubic ins., per second, while at atmospheric pressure the volume would be more than 10 times as great. But even 144 cubic ins. of steam per square foot of water surface is a large quantity, amounting to about two bubbles 1 in. in diameter per second per square inch of surface, or 2,000 per second if only $\frac{1}{10}$ in. in diameter. No wonder, then, if the water contains any frothing substances, that the bubbles will not burst till they are carried into the steam pipe, particularly as it takes less than forty seconds to remove all the steam contained in the steam space.

FIG. 91.

It is well known that it is quite impossible to make soap bubbles in an electrified atmosphere, and it is very surprising that in the days of low pressures no attempts were ever made to turn this knowledge to practical account.

Attempts to measure the amount of priming water were first made by Professor Thurston ('Am. C. E.,' 1874), by blowing part of the steam intended for the engines into a calorimeter, and measuring its specific heat. Comparing it with Regnault's results, the amount of suspended water could be calculated. This method being too complicated, the author, while at sea, tried to determine the amount of salt carried over by the steam, first by means of a salinometer, and then by means of the chloride of silver test, but was unable to detect any. During the trials of the Research Committee ('M. E.,' 1890) and later C. J. Wilson has, with the help of a more delicate test, been able to measure the priming water with great accuracy, and in all future trials of boilers the tests, which are very simple, should be repeated. They can easily be carried out at sea, and are also of special value to determine whether the condenser is leaking or not. To a measured sample of condensed steam a drop of a solution of yellow chromate of potash is added, and a weak solution of nitrate of silver—say 1 per 1,000—is slowly added, until the water suddenly turns brown (fig. 91). Repeat the experiment with a much smaller sample of water drawn direct from the boiler. It may require relatively from 20 to 100 times as much nitrate of silver as in the last case, and the ratio between the two is the ratio of priming water to steam. When testing the condenser water, in order to ascertain whether there are any leaky tubes, the comparison must be made with sea and not

with boiler water. As a check on these results add to the condensed water as much boiler water as, according to the test, was primed over, and repeat the experiment. The amount of nitrate of silver should then, of course, be twice as much as that used in the first test. The water which was drawn from the boiler has been partly evaporated by its own heat. The correction is made by ascertaining the temperature of the steam with the help of any reliable table. Subtract it from 1,178° F., then divide by 966, and multiply the previously found percentage of priming water by this quotient.

Great care has to be taken that no solid salt is contained in the brine cock, as this would materially affect the result; and it is also of importance to draw the steam out of the main steam pipe as near the stop valve as is possible, otherwise the priming water will have settled on the walls. For rough estimates it will suffice if the drain water from the high-pressure valve chest is taken. When the priming is to be measured of a marine boiler which uses no sea water, it is necessary to add a little pure salt—about $\frac{1}{3}$ oz. per gallon of water, or 5 lbs. per ton. This quantity is sufficient, yet it is so small that it cannot possibly do any harm. The priming of non-condensing engines can be measured in a similar way. A little salt is first added, and then at stated intervals water is drawn out of the boiler, and if there is any loss of salt, this is the measure of the amount of priming.

The following table has been calculated from Dr. H. Landholt and Dr. R. Bornstein's tables (1883.)

Saltness of Water at Saturation Point.

Temperature		Amount of Salt				Density when Cold
		Added to Water	Contained in the Water			
° F.	° C.	%	%	Oz. per Gallon	$\frac{1}{32}$	
32	0	35·5	26·2	50·3	8·4	1·2027
50	10	35·8	26·3	50·5	”	1·2036
68	20	36·0	26·4	50·7	”	1·2045
86	30	36·3	26·6	51·1	8·5	”
104	40	36 6	26·8	51·4	8·6	”
122	50	37·0	27·0	51·8	”	”
140	60	37·2	27·2	52·2	8·7	”
158	70	37·9	27·4	52·6	”	”
176	80	38·2	27·6	53 0	8·8	”
194	90	38·8	27·9	53·6	8·9	”
212	100	39·6	28·3	54·4	9·0	”

Sea water contains about 2·7 % of common salt and a few other soluble ones, making a total of $\frac{1}{32}$ of its weight.

CHAPTER II.

CORROSION.

EXPERIMENTS and papers on this subject are fairly numerous, and before discussing the various theories it will be advantageous to mention them. R. Mallet, 'Brit. Assoc.,' 1838, vol. viii. p. 253; 1840, vol. x. p. 221; 1843, vol. xiii. p. 1; and 'N. A.,' 1872, vol. xiii. p. 90.—In these experiments and papers the question of boiler corrosion is hardly touched upon, but galvanic action and related subjects are thoroughly discussed.

'Parliamentary Reports,' Admiralty Committee on Corrosion in Boilers, appointed in 1874; three reports, 1874, 1878, and 1880, c. 2662. —These experiments were very exhaustive; they aimed at ascertaining whether there was a difference between the behaviour of iron and steel, whether the lubricants in the engines affected boiler corrosion, and at determining the influence of zinc, of galvanic action, of various fluids, and of air in water.

D. Phillips, 'C. E.,' 1881, vol. lxv. p. 73, discusses the above reports, and comes to the conclusion that iron does not corrode as fast as steel. W. Parker, 'I. and S. I.,' 1881, p. 39, like the Boiler Committee, exposed various materials in actual boilers, but isolated each plate. The results do not show a great difference between iron and steel. D. Phillips, 'Marine E.,' 1890-91, adds further experiments in support of the above-mentioned views.

Other experiments on corrosion will be found in the following papers:—A. Mercier, 'An. Mines,' 1879, 7th ser. vol. xv. p. 234, gives experiments on the influence of fatty matter on corrosion of iron and steel. M. Lodin, 'Comp. Rend.,' 1880, vol. xci. p. 217, experiments on corrosion of various wires in hot fluids to which vegetable matter has been added. M. B. Jamieson, 'C. E.,' 1881, vol. lxv. p. 323. J. Norris, 'N. A.,' 1882, vol. xxiii. p. 151, determined the influence on corrosion of the air contained in water. L. Gruner, 'An. Mines,' 1883, 8th ser. vol. iii. p. 5, relative corrosion of 28 materials under 4 different conditions. This paper contains an appendix by Bustein, which shows that exposures of the samples for 112 days inside a boiler, and also in a boiler flue, affected the mechanical properties (see p. 113). J. R. Fothergill, 'M. E.,' 1884, p. 339; F. Marshall, p. 344. Professor Lewes, 'N. A.,' 1887, vol. xxviii. p. 247, influence on corrosion of the air contained in water, and general views on corrosion, chiefly in sea water. A. C. Brown, 'I. and S. I.,' 1888, ii. p. 129. Professor Lewes, 'N. A.,' 1889, vol. xxx. p. 340. T. Andrews, 'C. E.,' 1884, vol. lxxvii. p. 323, and 1885, vol. lxxxii. p. 281, gives experiments on corrosion in

sea water. A. Wagner, 'Dingler's J.,' vol. ccxviii. p. 70, various theories on the chemistry of corrosion. Bursteyn, 'Mitt. Pola,' 1879, vol. vii. p. 503, experiments on the influence of high pressure and fatty matter. H. Schnyder ('Berg.- H.-Z.,' vol. xxvii. p. 212), experiments on the behaviour of zinc under various conditions.

The main object of all these experiments is to ascertain the true causes of corrosion, and to discover means for preventing it; and since steel has been substituted for iron the question of its relative liability to corrode has repeatedly come to the front. No engineer with extended experience will hesitate to admit that steel does behave worse than iron. Fortunately this view has not only not led them to return to iron, with its numerous bad properties, but it has driven them to adopt preventives which have now reduced corrosion in both iron and steel boilers to very small proportions. These preventives are—

I. The substitution of mineral lubricants for animal or vegetable fats or oils.

II. The use of fresh or even distilled water wherever obtainable instead of sea water.

III. The removal of air from the feed water.

IV. The use of zinc.

V. The use of alkalies.

In order to understand how these practices have been arrived at it will be necessary to discuss the various theories of corrosion.

I. and V. **Action of Engine Lubricants.**—All vegetable and animal fats or oils behave chemically as if they were salts, consisting of a base, and an acid. In fats and oils the base is glycerine, and the acids have numerous names. Fats and oils, like some other neutral compounds—for instance, the carbonates of lime and magnesia and the sulphate of iron—are split up into acids and bases when heated. The temperature at which this splitting up takes place with fats is a little above 212° F. Thus stearine, which is the same substance as stearic acid, is manufactured by raising tallow and water to the proper temperature. Naturally the same thing can happen in a boiler, and the liberated fatty acids are now capable of corroding the same amount of iron as an equal weight of sulphuric or other mineral acid. The resultant compound is ferric soap, which makes up the greater part of the filthy greasy substance to be met with in some boilers. By bringing the fatty acids in contact with other substances, other soaps are formed. Thus with potash we get soft soap, with soda hard soap; with lime we get the very much harder soap called putty, while with the oxides of lead and zinc medical ointments are the products.

Whereas glycerine is one of the weakest bases, lime, potash, and soda are the very strongest, and whereas the one is incapable of retaining the fatty acids at a boiling temperature, the others will only part with them in presence of a stronger acid. Carbonic acid is one of the weaker ones, and therefore it is possible for the acids in some oils and fats to supplant it, if brought into contact with carbonate of soda. Some of them are not strong enough to do this direct, but must first attack the iron, forming ferric soap. In such a case the soda does not prevent corrosion, while in the previous one the liberated carbonic acid

might do just as much harm as the fat. When using organic lubricants it is, therefore, better to add caustic soda or lime instead of the carbonate. Mineral oils are not double compounds, and, like water, consist of only two elements—viz. carbon and hydrogen—and these are harmless. To add soda where these are in use would be useless.

II. **The Use of Fresh Water** is advantageous, because it contains no corrosive elements. Nor should the addition of neutral salts produce any corrosion; but chloride of magnesia, which is always present in sea water, is a very important exception, being exceedingly injurious. A. Wagner shows that if water contains chloride of magnesia, but no air, it commences to attack iron at a temperature of about 212° F., while the following chlorides will only attack it in presence of air :—they are arranged in the order of their corrosive power—ammonium, sodium, potassium, barium, calcium.

The Boiler Committee tube No. 21 contained chloride of magnesia, and attacked iron and steel very severely.

Professor Lewes states that if sea water is distilled while in contact with iron it gives off hydrochloric gas when the volume of the water has been reduced to one-fifth, and it is only too probable that before it is produced in sufficient quantities to escape it must have been attacking the iron. He also mentions that magnesic chloride and calcic carbonate (lime) react on each other, and are converted into calcic chloride and magnesic oxide, the carbonic acid escaping at a boiling temperature. When exposed to the influence of the air magnesic oxide absorbs carbonic acid, so that on re-filling a boiler which has been open for some time, and heating it, this acid gas is given off again. Magnesium salts must, therefore, be looked upon as decidedly injurious to the life of a boiler; but, as all sea water contains them, this should, if possible, never be admitted.

Free Acids, which are sometimes proposed for boiler-scale solvents, should never be used, nor should the feed water ever be taken from a river near a chemical factory. Occasionally waste acids are discharged there, which may do a serious amount of injury. This was mentioned by Mr. Hallett ('M. E.,' 1884, p. 350).

Whenever there is any doubt as to the harmlessness of fluids or salts intended to be put into a boiler, a simple test is to boil them, and then to place a thoroughly clean knife-blade into them. Should any rust be formed, should the water be discoloured, or should copper deposit itself on the blade, then the substance ought not to be used. If certain free acids are present the above test will give no warning, but a few drops of dissolved yellow prussiate of potash or of tannic acid should be added. If iron has been dissolved, a light bluish precipitate is at once formed by the first test, which slowly turns dark blue. The other fluid produces ink.

Copper Salts seem to be an unavoidable, though only a minute, constituent of all feed water, as is proved by the green scale in all old boilers. For some unaccountable reason it does not deposit itself uniformly over the inside of the boiler. Its presence is denied by many engineers, therefore little information could be obtained as to the points where it is most generally found. The author's experience is that patches of green scale will be found on the zinc slabs, and also near them on the iron. This is true even when the zinc *t* is suspended

from the steam-space stays (fig. 92). It will be found that the tubes marked + sometimes contain thick patches of greenish-grey scale. Small quantities are also found near the water lines of boilers, particularly at both end plates. Larger quantities are found at the front end plates, between the nests of tubes, and on the lower part of the shell plate. The furnace bottoms usually contain the greatest number of patches, but it is difficult to detect them, as these parts are discoloured by grease. The furnace crowns and combustion chambers are nearly always quite free from them, from which it would appear that it is chiefly the non-heating surfaces of a boiler which get covered.

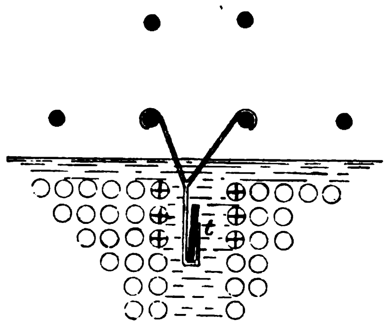


FIG. 92.

If the copper would only distribute itself uniformly, it would act as a protective scale, but as it is deposited on small areas, these may become sources of danger by producing galvanic currents, at any rate if there are any powerful chemicals in the boiler water, either acids or alkalies, or even neutral salts.

Unless copper pipes are fitted inside the boiler, the only sources of supply are the condenser tubes, the feed pipes, and the pumps. Here again it is the use of vegetable and animal lubricants which causes the mischief, for both readily attack copper or brass, as can be seen by their greenish colour if either has been in contact with these metals. Distilled water also seems to be a solvent; it certainly attacks lead. It is also believed that the minute particles which are worn off the working parts of the pumps are carried bodily into the boiler, and J. MacFarlane Gray ('N.A.,' 1861, vol. ii. p. 157) has detected specks at the bottom of pit holes, and attributes pitting to that cause; and Professor Lewes (' N. A.,' 1889, vol. xxx. p. 340) also believes that the presence of copper causes pitting. The author's observations do not support either view, for although he has been very careful to observe green scale patches, he never could detect signs of pitting near them, nor did the corrosion seem in any way to be increased at these points. These views are supported by Mr. H. W. Hirman's statement, in answer to Mr. MacFarlane Gray (see above), that land boilers using town water pitted, although here there could be no question of the presence of copper. Additional remarks on galvanic action will be found further on.

Zinc Salts.—There remains one set of salts which possibly play an important part in boilers of the present day, but of which no mention is made in any books or papers, and the following remarks are therefore purely speculative. It is a well-known fact that if boiler water is often renewed the reduction of the zinc slabs is more rapid than if the same water is used over and over again. A very obvious explanation would be, that the more zinc salts are dissolved in the water, the less corrosive it is. If this view is correct, the painting of the insides of

boilers with zinc oxide, and the addition of some zinc salts—the chloride of course excepted—to the boiler water, ought to have a beneficial effect, and recent limited experiences show this to be the case.

Neutral Salts.—The following experiments throw a little light on the part played by what would appear to be perfectly harmless salts in increasing the corrosive power of acids ('Journal of the Camera Club,' London, 1892, vol. vi. p. 52). M. Gourdon's experiments in 1873 show that exceedingly diluted sulphuric acid could still be made to attack zinc metal by adding to it various salts.

Added Salts of	Dilution of Acid in Water
Cobalt	1 : 10,000
Nickel	1 : 7,000
Platinum	1 : 7,000
Iron	1 : 7,000
Gold	1 : 5,000
Copper	1 : 4,000
Silver	1 : 3,500
Tin	1 : 1,500
Antimony	1 : 700
Bismuth	1 : 500
Lead	1 : 400

On the same page L. Warnerke gives a table showing that, by adding various salts to a one per cent. solution of sulphuric acid, the speed with which it attacks zinc metal can be varied considerably. The amount of corrosion is given in decimals of millimeters of depth per hour.

Salts added to the Diltue Sulphuric Acid	Speed of Corrosion
Nickel ammonio-tartrate	·13 mm.
Cobalt chloride	·11
Iridium chloride	·09
Palladium chloride	·085
Nickel cyanide	·077
Chrom. chloride	·070
Gold chloride	·070
Silver ammonio-nitrate	·025
Lead nitrate	·010

The interesting point about these results is, that both lead and copper salts stand very low down in the two lists, showing that the accepted notion, that they are very injurious, is, in one sense at least, a wrong one; and the bare fact that these various salts can influence the behaviour of sulphuric acid makes it appear probable that zinc salts also have an influence on the action of corrosive substances found in a boiler—apparently a beneficial one.[1] Possibly experiments carried out on the above lines might show that manganese salts, which enter the boiler water as the steel plates corrode, act injuriously, and this might explain why the presence of this metal in steel has been looked upon as increasing its corrodibility.

Alkalies.—The other substances which find their way into a boiler

[1] Mercury salts are known to be efficient protectors against rust.

are salt and lime, and in some ships slacked lime, caustic potash, or soda; their carbonates are also sometimes added. It cannot be said that much is known about their action, except that any one of the above alkalies will neutralise any free acids. When added in excess they too might do harm, particularly if copper is present in a boiler, as they readily produce galvanic currents. (See A. Wagner.)

III. **Air in Boilers.**—The action of the air, which generally accompanies feed water, is discussed in the papers which have been already mentioned. A. Wagner found that pure water will not attack iron, except in the presence of air; that the action is more severe if carbonic acid is present, and still more so if the chlorides of magnesium, ammonium, sodium, potassium, barium, or calcium are present, the magnesic chloride being the most powerful. These views are corroborated by most of the other writers.

It would seem as if pitting may safely be attributed to the presence of absorbed, or, as it is generally called, occluded oxygen, or to carbonic acid, or to both.

A certain form of surface defects of boiler plates, due to hard lumps of slag having been rolled into them, should not be mistaken for pitting. The cavities, which often make their appearance, are due to the boiling out of the slag, but this only happens after the boiler has been in use some time. These cavities are slightly elongated in the direction of the fibre, and sometimes look like a series of irregular waves. The solution of this scale produces the black and the red mud found in new high-pressure boilers.

The Absorption of Air by a fluid is stated to take place as follows: The volume of gas which a definite quantity of fluid can absorb is independent of the pressure, and decreases with rising temperature. Therefore, as 1,000 cubic ins. of water of 0° C. will absorb 41·15 cubic ins. of oxygen at atmospheric pressure, they will also absorb an equal volume at two atmospheres, and so on. Of course, as the pressure is increased the density of the oxygen is increased, so that the *weight* of absorbed gas is proportional to the pressure. This law also holds good for a mixture of gases.

As an example, estimate the quantity of air which pure cold water will absorb. The coefficients are ·04115 for oxygen, ·02035 for nitrogen. Their respective densities are as 16 to 14. And as there are 20 % of oxygen and 80 % of nitrogen in the air, the densities of these two gases are as $\frac{1}{5}$ and $\frac{4}{5}$ respectively of what they would be if pure. 1,000 cubic ins. of water will therefore absorb 41·15 cubic ins. of oxygen, having a density of $\frac{1}{5} \times 16$, and 20·35 cubic ins. of nitrogen, having a density of $\frac{4}{5} \times 14$. The relative weights of the absorbed gases would therefore be as $41{\cdot}15 \times \frac{1}{5} \times 16 = 124{\cdot}5$ to $20{\cdot}35 \times \frac{4}{5} \times 14 = 228{\cdot}5$, or as ·55 to 1·00. In the atmosphere their relative weights are as ·286 to 1·000. This shows that by compressing air in presence of water, and then allowing it to escape again, a mixture would be obtained in which there is about 100 per cent. more oxygen than in the atmosphere. Practical difficulties stand in the way of utilising this action commercially.

Air contained in Hot Boiler Water.—A similar calculation will show how much oxygen is contained in boiler water. If, as is not impossible, the steam should contain one-tenth per cent. of oxygen, and

if its pressure is 150 lbs. (10 atmospheres), then the density of the oxygen is one-hundredth of what it would be if pure and if subjected to a pressure of one atmosphere. Every thousand cubic inches of water, if cold, would therefore absorb as much oxygen as would measure ·4115 cubic in. under atmospheric pressure, or one-twentieth of what is found in fresh water; but taking into account that the temperature is high, that the coefficient of absorption is probably much smaller than when cold, and that the density is also less, there can be little doubt that only traces of oxygen are to be found in the hot water of a boiler.

In the same way that silicon and aluminium expel air out of molten steel, some salts may be expected to drive it out of water, but little is known on this subject (see p. 105, also M. Lodin, 'Comp. Rend.,' 1880, vol. xci. p. 217).

Air in Feed Water.—The conditions under which air and water meet in the feed pumps and their air chambers are totally different from the above. There the temperature is low, and the pressure of the steam vapour so slight that it can be neglected. During each delivery stroke the pressure on the air and water is raised to 150 lbs., which would cause 8 % volume of oxygen and 16 % of nitrogen to be absorbed, i.e. 3·2 lbs. of oxygen per ton of cold feed water. That there is nothing improbable about this is clearly demonstrated by the fact that Sir W. Thomson's (Lord Kelvin) sounding apparatus cannot be used for great depths, because at 200 fathoms (= 40 atmospheres) all the air in the glass tube will be absorbed by the water which has entered it.

Position of Feed Discharge.—Having admitted the saturated feed into the boiler, there are three ways of dealing with it:—

1. Either lead it so that it enters the bottom of the boiler or easily falls there.

2. Admit it at some point in the boiler—say, over the back end of the tubes, so that it gets thoroughly mixed with the hot water and loses its air.

3. Lead it through pipes which are fixed inside the boiler, in order that it may become sufficiently heated to part with all its air.

All these plans are in use. In the case of No. 3 the internal pipes, if made of iron, suffer very seriously from pitting. If made of copper they also suffer; and in such cases much green scale will be found in the boiler. The joints inside the boiler should not be cemented, otherwise water-hammer action takes place after a short stoppage of the feed, and bursts the pipes.

No. 2 is the general practice. Few engineers have had the courage to discharge into the steam space, but where tried the plan works satisfactorily, and ought to assist the water circulation, on account of 20 % more steam being generated and condensed. Boilers liable to prime could not be worked in this way, as was mentioned by J. H. Hallett ('M. E.,' 1884, p. 350); but at any rate the feed should then be introduced at a point where it will be heated as quickly as possible.

No plan could be worse than to discharge the feed at the bottom of a boiler, for it is clear that, in comparison with other parts, the spaces under and between the furnaces can only have a very restricted circulation. This defect is increased if they are filled with

cold, and therefore heavy, water (see fig. 93: the dark spaces at the bottom represent cold water); then, as the fire bars, the ashes, and

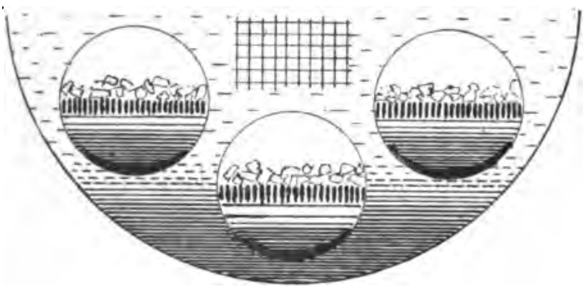


FIG. 93.

the inrushing cold air all prevent heat from reaching the bottoms of the furnaces, the water has absolutely no chance of rising except by the very slow process of being fed from below. This speed is about 5 ft. per hour, or 1 in. per minute. There is, therefore, no difficulty in heating all the feed slowly and along one particular zone, viz. along the line of the fire bars. Under these conditions the straining to which both the shell and furnaces, but particularly the latter, are subjected, must be excessive, for not only is there a sudden jump from cold air below the grate to white heat above them, but on the water side there is also a sudden rise from about 100° F. to over 350° F.

Pitting.—It is, however, not with stresses and circulation that we are at present dealing, but with corrosion, due to air absorbed by the feed water. What takes place with this air is shown in fig. 94. As soon as the cold water comes in contact with the warm part of the furnace plate, F, it is heated and compelled to give up its air, and being in contact with the plate, the air settles on it. There being no circulation, it is only when the bubbles have grown sufficiently large that they rise. But during this period of rest the nascent oxygen which they contain will attack the iron, and having formed small irregularities, subsequent bubbles find a still better lodgment and speedily effect the formation of pit holes. If the feed is led into the bottom of the boiler, and if it is saturated with air, it can be shown that every inch of furnace length generates about four and a half cubic inches of air per hour. This is equal to about one bubble $\frac{1}{8}$ in. diameter per second. The excessive differences of temperature along this line of

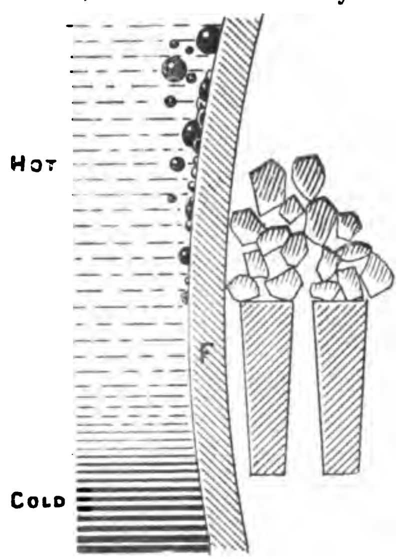


FIG. 94.

grate, and the consequent excessive straining of these parts, quickly loosen all rust as it is formed, so that metallic iron or steel is always exposed to the air if allowed to be produced there. Certain it is that if there is any pitting going on in a boiler, the greater part is sure to be found along the line of fire bars. Another part which is also severely attacked is the under side of the furnace and combustion chamber, for here the air bubbles cannot rise if they have once been formed.

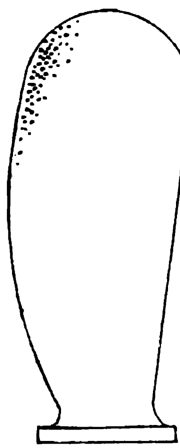


FIG. 95.

In support of these views it may be pointed out that severe pitting frequently takes place on the forward side of propeller blades at the leading edge, as shown in fig. 95. Here the air is not driven out by heat, but is abstracted by the partial vacuum which is found there, and, in spite of the high velocity of the water, there seems to be sufficient time for the mischief to be done. Even bronze blades are sometimes pitted at this point.

Another confirmation is found in the severe pitting of internal iron feed pipes; even copper ones waste away. Here the air is liberated by the transmitted heat of the surrounding water, and it has been suggested that long internal iron feed pipes should be fitted to boilers, and renewed whenever they are eaten through, for, whether this is due to the air or some other corrosive agent, it is cheaper to lose a regular quantity of temporary piping than to have to renew furnaces. They last about eighteen months.

Distribution of Corrosion.—One peculiarity about corrosion in general is that boilers which suffer much at their lower parts are often found to be as good as new in their steam spaces. In others the stays and plates of the steam space suffer severely, while the lower parts are only slightly attacked. This happens chiefly when the feed is discharged near the water level, but, as it is next to impossible to say in what path the circulation in a boiler takes place, and whether the feed is carried up or down, definite conclusions cannot be drawn. Another curious fact is, that if a number of boilers are connected to a single feed pipe, that one which is farthest away from the pumps suffers more than the others.

Steam-Space Corrosion.—If it is the air which attacks the steam-space stays and superheater plates, there is perhaps no other remedy than not to admit it, unless it can be proved that zinc will here also act as a protector. It seems to do so, but this can hardly be through galvanic action. Possibly here again it is the presence of zinc salts, of which it is not difficult to imagine that they have been carried into the steam space. In some steamers these parts are whitewashed with zinc white, and the result seems to be a satisfactory one.

It is of course possible that this corrosion is due to steam alone, or to the hydrochloric acid which, as has been mentioned, escapes from sea water when evaporated in iron boilers; but it is safer to blame the air, as it has not yet been shown that steam can attack iron at temperatures ranging near the boiling point of water. If it did, the action

ought to be equally strong in all boilers, and that is far from being the case. That this form of corrosion may lead to serious results is only to be expected, because of its rare occurrence and on account of the wasting being very uniform. Thus, rivet heads and flat plates in steam spaces retain their original shape even when most of their substance is gone. As an instance a case may be mentioned where the rivet heads of a steam dome were corroded as shown (fig. 96). The head seemed to be intact, but had in reality disappeared, and several of the rivets could be driven back with a hand hammer. The dotted line shows the original thickness of the plate and size of rivet head.

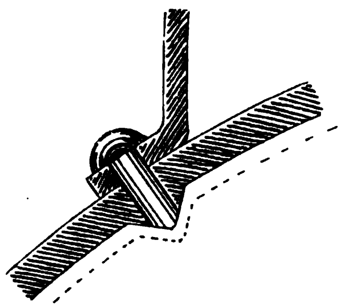


FIG. 96.

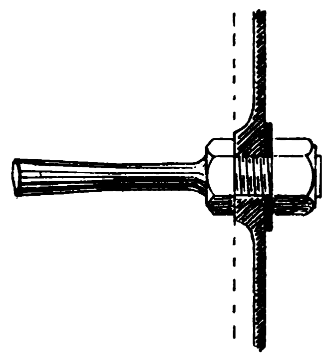


FIG. 97.

In another case the steam-space end plate of a boiler was wasted as shown (fig. 97), and it was only due to the necessity of renewing the stays that the corrosion of the plate was discovered, for the shoulder round the nut had been mistaken for the washer which is sometimes placed there. The wasting of the front plates and front ends of the steam stays is aggravated by the heat of the uptake; but, as equally bad cases are met with when baffle plates are fitted, the temperature cannot be the sole cause.

The following is a curious illustration of this sort of corrosion. On examining a pair of boilers of a ship which was barely four years old, it was found that, although efficient baffle plates had been fitted, five of the steam-space stays, situated about 2 ft. above the water level, had lost nearly 30 % of their substance, but only close to the smoke-box end. The starboard boiler was in a perfect condition. Zinc had been used in both, but had been given up. There was only one difference in the structure of these boilers, which is so small that it would not be worth mentioning were it not in connection with the feed. It was found that the knee pipes, K (fig. 98), which had been screwed to the feed inlets, in order to produce downward currents, had fallen off in the port boiler, and therefore a possibility exists that the feed water of that boiler travelled in the direction shown by the arrow F, and would part with its air sooner and deliver it into the steam space.

A most salutary lesson was taught some years ago, when the admixture of air to the boiler feed was advocated as an improvement to the circulation. Its injurious effect was so great that the idea very soon died out. The Boiler Committee also helped to convince engineers on this point, and in all high-class engines care is taken to keep the air out of the boiler, this being the simplest and safest plan for guarding against its corrosive action.

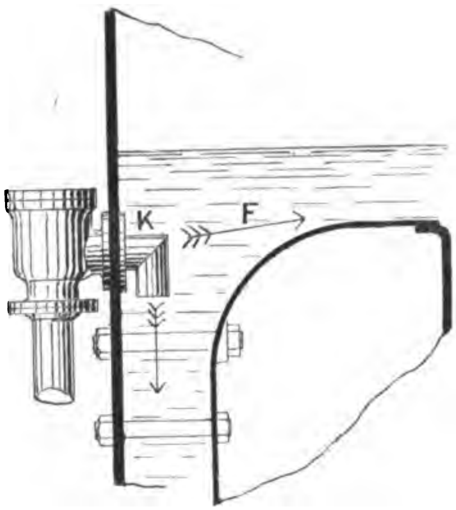


FIG. 98.

Separate automatic feed pumps are the only efficient means of securing this object.

Galvanic Action.– The laws which have been discovered in this branch of science may be briefly stated as follows: A continuous electric current is only possible in a circuit. Its intensity, measured in ampères, is equal at all points of this circuit, and is proportional to the algebraical sum of the electromotive forces in the circuit (measured in volts), and inversely proportional to the sum of the electrical resistances of the circuit (measured in ohms).

The electromotive force makes its appearance under various circumstances, viz. when two different substances are brought into contact, or when changes of electricity or magnetism take place near the circuit, or when heat is made to travel along part of a circuit. Here only the former case will be dealt with.

If a circuit is constructed consisting only of solids, then no current will be generated, because the sum of electromotive forces equals nought. Therefore, if the electromotive force due to contact of one metal with several others is known, that between two of these is found by the difference. The following example will illustrate this.

The electromotive forces in volts due to contact with iron are—

Carbon	Platinum	Copper	Iron	Tin	Lead	Zinc
–·485	–·369	–·146	0	+·313	+·401	+·600

To find the value when, for instance, carbon and copper are brought into contact we have $-\cdot485-(-\cdot146)=-\cdot339$. These values change considerably with rises of temperature. If, therefore, one of the points of contact in a metallic circuit is heated, the electromotive force at this point will be altered, and a current produced. Thus, at 530° F. the electromotive force between copper and iron is reduced to 0, and a circuit consisting of only these two metals,

but with one joint heated and the other cold, would show an excess of potential of $-\cdot 146$ volt in one direction. All thermo-electric piles are constructed on this principle. With fluids, or with solids and fluids, this simple law does not exist, as will be seen from the following values of electromotive force which appear on immersing any of the above solids in pure distilled water, and also in sea water.

Table.—Electromotive Forces.

Fluids	Carbon	Platinum	Copper	Iron	Tin	Lead	Zinc
Distilled water	From $+\cdot 010$ to $+\cdot 017$	$+\cdot 285$ / $+\cdot 345$	$+\cdot 100$ / $+\cdot 269$	$+\cdot 148$	$+\cdot 177$	$+\cdot 171$	$-\cdot 505$ / $+\cdot 156$
Sea water	?	$-\cdot 856$	$-\cdot 475$	$-\cdot 605$	$-\cdot 334$	$-\cdot 267$	$-\cdot 565$

The electromotive force in the circuit (fig. 99) would therefore be as follows:—

Contact copper and iron . .	$-\cdot 146$ volt
,, iron and salt water .	$-\cdot 605$,,
,, salt water and copper	$+\cdot 475$,,
Electromotive force in circuit .	$-\cdot 276$ volt

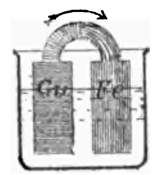


FIG. 99.

A Daniell's cell, consisting of copper, zinc, sulphate of zinc, porous cell, sulphate of copper, has an electromotive force of about 1·10 volt.

One effect of a current, circulating in a circuit, is to generate heat at every point, the amount being proportional to the square of its intensity and to the local resistance.

Thus, if the entire resistance of a circuit is R ohms, its electromotive force E volts, and its intensity I ampères, we have—

$$I = \frac{E}{R}, \text{ and } W = I^2 . R = \frac{E^2}{R} = E . I.$$

Here W is the amount of work done in the whole circuit measured in *watts*, of which 746 are equal to one horse-power and 1,049 for one second are equal to 1 heat unit.

Another effect of the electric current is to produce chemical changes in the fluids through which it passes, usually splitting them up into their elements. It has been found that the amounts are strictly proportional to their atomic weights. Thus a current of one ampère, passing through water during one second, will produce ·0001038 gram of hydrogen, and $2 \times 15{\cdot}96$ times as much oxygen, viz. ·0008283 gram. If the electricity was produced by a Daniell's cell, which then of course forms part of the circuit, it would be found that during this second its zinc has lost ·003367 gram, and its copper gained ·003279 gram, these quantities being proportional to their respective atomic weights. Iron would have lost ·002900 gram.

This loss of metal at the electrodes is a secondary action, being due to the dissolving power of the elements produced there. Oxygen and hydrogen would be liberated from salt water ; but the former gas, being nascent, would combine with the iron, if that is the metal of the electrodes.

It is evident that the chemicals produced by these secondary actions must influence the electromotive forces in the circuit; this is called polarisation. Iron shows this property in a very marked degree.

If the positive electrode is made of iron, which, as has been stated, is a distinctly electropositive metal, it will very soon change its nature, and grow even more electronegative than copper. The unexposed parts will still be electropositive. If the whole piece of the iron is now placed in the fluid, a strong current is set up from its negative to its positive end, and through the fluid back to the negative part; but very soon the current ceases, and on examination it will be found that the electronegative property has spread over the whole piece. There are other means for making iron electronegative, viz. dipping it into concentrated nitric acid, or heating it in air—in fact, anything that will oxidise it; and there seems little doubt but that this form of polarisation is due to a scale of oxide of iron, which in some cases is so fine as to be invisible. This would lead to the conclusion that iron ought not to rust beyond its initial stage; but as it does so, particularly in boilers, there is no alternative but to admit that this beneficial change is not possible at a boiling temperature, or that the galvanic action, which produces it and causes it to spread, does not exist in a boiler, or will not act in the same way on large surfaces as on small ones. Electronegative iron can be brought back to its primary condition by making it the negative electrode. This is probably due to the reducing action of the hydrogen on the oxide of iron.

The presence of hydrogen on the electrodes also influences the workings of a circuit, partly increasing the resistance and partly making the electrode more electropositive. It is, therefore, not surprising that the galvanic actions either do not occur, or remain unnoticed when the electromotive forces are weak. Some allowance should therefore be made for the density of a current; this is measured by dividing its intensity by the sectional area of any particular point of a circuit; and it is of interest to note that the denser it is at the negative electrode, the more ozone instead of oxygen is generated there.

Galvanic Action of Black Scale.—As regards the possible electric actions inside a boiler, there are currents passing from the exposed iron through the water to patches of slag and back, tending to dissolve the iron and to make it as electronegative as the slag, when all action ought to cease.

Galvanic Action of Zinc. If slabs of zinc are fitted in such a manner as to be in metallic contact with the iron (see fig. 100), a current passes from zinc to water, to iron, and back, zinc being dissolved and hydrogen evolved on the iron surface.

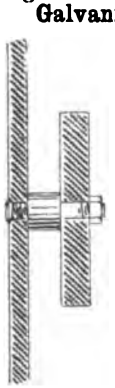


FIG. 100.

If the zinc is connected to the iron plates by means of a copper wire, there will be two circuits, viz. one from the zinc to water, to copper, and one from iron to water, to copper; the one current wastes the zinc, the other the iron. If, therefore, the zinc does protect the iron from corrosion, it must be due to some other cause than galvanic action. Copper pipes should produce a current from iron to water, to copper, and the iron should grow electronegative.

The intensities of electric currents which might be expected in a boiler, if no polarisation took place, are very difficult to estimate, as the laws which govern the flow of electricity in conductors of large dimension have not yet been clearly formulated. The resistance of a conductor is proportional to its length, and inversely proportional to its sectional

area. For a cubic centimeter and per cubic inch we have the following resistances measured in ohms :

	Per Cubic Centimeter	Per Cubic Inch
Silver	·000,001,579	·000,000,621
Copper	·000,001,611	·000,000,635
Iron	·000,009,638	·000,003,81
Mercury	·000,094,340	·000,037,1
1 water + ·055 salt	91·2	35·9
1 „ + ·0425 „	123·0	48·5
1 „ + ·0212 „	235·0	92·6
1 „ + ·0106 „	434·0	170·8

This table shows that the resistance in the metal may be neglected, but it is also evident that, except when the electrodes are almost in contact, the fluid resistance will reduce the intensity of the current excessively.

Take, for instance, a strip of iron 1 in. wide (fig. 101), from which part of the scale, S, has been removed, exposing the iron at Fe to the salt water, W ; then a current will flow from Fe to W, to S, and back. Restricting the observation to the small zone of the diameter D, the available section of the conductor is $\frac{D}{2}$—say, $\frac{1}{32}$ in.—and its mean length from Fe to S is $\frac{1\cdot507}{32}$ in. Assuming that the water contains 5·5 % salt, the resistance of this small element would be (35·9 × 1·507 =) 54·2 ohms + the resistance of the scale and iron, say 60 ohms.

Assuming also that the electromotive force of iron scale in salt water is the same as for copper, the electromotive force of such an element would be ·276, which would give a current of $\frac{\cdot276}{60}$ = ·0046, which would be capable of corroding ·0022 lb. per day, or at the rate of about $\frac{1}{16}$ in. in 60 days. At one inch distance from the edge of the scale (see fig. 101) the action would be about 3 % of the above, showing how rapidly it diminishes with the distance.

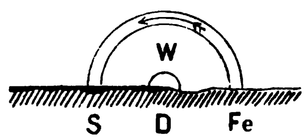


FIG. 101.

Pitting has been explained in this way, and this view is to a certain extent corroborated by the experiments of Mr. Parker, who found that the average depth of corrosion in plates from which the scale had been only partly removed amounted to 20 % more than that of the bright discs. But then their average losses were not sufficiently uniform to permit of strong conclusions being based on them.

The above calculations will also make it clear that the necessity for isolating various samples while subjected to experimental corrosive influences does not exist, and therefore the very exhaustive experiments of the Admiralty Boiler Committee are worthy of more confidence than is generally placed in them.

The Galvanic Action of Zinc in Boilers.—If galvanic currents exist, the studs by which the slabs are secured ought to remain bright, at least within one-sixteenth of an inch of the metal ; but it is well known to those who take much trouble in keeping up a good metallic connection that corrosion takes place even at the contact surfaces, which have to be scraped or filed after every voyage. The current produced by zinc, iron, and sea water is about twice as strong as that of scale and iron, but as it has no effect on its immediate surrounding,

it is hardly probable that it will have much influence on the more distant parts, which are separated from it by several feet of salt water.

Composition of Boiler Plates and Corrosion.—A subject to which sufficient attention has not yet been paid is the influence of the composition of iron and steel on their behaviour in the presence of corrosive fluids. It is believed that manganese increases corrosion, while there can be no question that nickel reduces it. Carbon behaves very strangely, and it would seem as if in one form—viz. as found in annealed steel—it increased corrosion, whereas in hardened steel it reduces it. However there is little scope for improvement, because its percentage is fixed by other considerations.

A few remarks on some of the results of the experiments carried out by the Boiler Committee will not be out of place here.

As in the case of Mr. Parker's experiments, small samples of steel and iron plates were exposed in various merchant and Government steamers, and their losses ascertained. The unit of comparison is 1 grain per square foot per 10 days, equal to about $\frac{1}{800}$ in. per annum.

It was found that when using jet condensers the corrosion was very erratic, ranging from 35·5 to 281·3 grains; but there were only four of these experiments

Influence of Condenser Tubes.—In four cases copper-tubed surface condensers were used, and the corrosion was slight. There were 25 cases of brass condenser tubes. The corrosion in their boilers was slight, except in two cases, with which respectively colza and Rangoon oil were used as lubricants in the cylinders. With these two exceptions the maximum and minimum losses were 128·8 and 16·8 grains. In only two of these cases was zinc used, and its influence was not marked. The cylinder lubricants were nearly always mineral oils. There were seventeen cases of tinned condenser tubes. Their influence was very marked, the corrosion being far severer than under the other conditions. Mineral oil was used in four of these ships, and the corrosion varied from 204·6 to 362·2 grains. In these cases the use of zinc alone does not seem to do much good, the losses in three cases being 129·6, 215·2, and 503·2 grains. On the other hand, when chalk, Portland cement, and, strange to say, tallow were placed in the boilers whose condenser tubes tubes were tinned, the losses were much reduced, two cases being as low as 32·7 and 39·7 grains. In two boilers soda was used. Its action is not a decided one, for the losses were 30·6 and 376·3 grains.

Consumption of Zinc.—In the reports of the Boiler Committee information will also be found as to the loss of zinc when fitted in boilers. In the case of H.M.S. 'Crocodile' 425 to 630 lbs. of rolled zinc were consumed per boiler per annum, while in H.M.S. 'Serapis' the quantity was 236 to 420 lbs. This is only the actual loss as found by weighing the zinc before and after each voyage. Analysis of zinc refuse in boiler will be found in 'Enging.,' vol. xlvii. p. 236.

In addition to the general remarks about preventing corrosion (p. 47) the following have suggested themselves. Strong chemicals, either alkalies or acids, should not be used. Certain salts increase or reduce the speed with which iron is attacked. Zinc salts appear to reduce the action, tin salts to increase it. Air, being the probable cause of pitting, should be carefully excluded. The existence of galvanic currents in boilers has not been proved.

CHAPTER III.

FUELS AND COMBUSTION.

Combustion.—When two substances, having a chemical affinity for each other, are brought together under favourable conditions, they combine, forming a new compound. During this combination heat is evolved, which, if the chemical affinity is sufficiently strong, will raise the temperature of the substances to such a height that they grow luminous. This is called combustion. The term is not restricted to the burning of coal, wood, oil, or gas in air, but is applicable to the burning of these or other substances in any other gas, such as chlorine or hydrogen, and is sometimes used to denote a similar process in which only solids or only fluids combine. It may be illustrated by igniting gunpowder, by dropping potassium metal into mercury, or by heating a mixture of iron filings and sulphur. In any of these cases luminous heat is generated.

Slow Combustion is a term which is commonly restricted to express decay of organic matter, but the process is met with on a large scale in every coal mine, where it is the cause of the increased temperature of the ventilating air as it leaves the shaft. In our lungs a process of slow combustion is continually proceeding, and in the tarnishing of metals the same action is manifested. Thus magnesium wire if lighted burns, if exposed to atmospheric influences it tarnishes. In both cases oxide of magnesium has been formed.

When burning coal in a boiler furnace, the main object is to obtain heat; slow and imperfect combustion have, therefore, to be prevented. The one takes place generally, but not always, if the air is in excess and too cold, while the other is caused by an insufficient supply.

Heat, as is well known, is not the same thing as temperature. It is a quantity and not a condition. It is measured in units of heat (called a calorie), of which each one will raise the temperature of one pound of water one degree Fahrenheit. It is often more convenient to use another measure, viz. the evaporative unit. This is equal to the amount of heat which will evaporate one pound of boiling water from and at 212° F. One evaporative unit equals 966 calories, or heat units, or thermal units (J. K. Cotterill, London, 1878, p. 314); it is also equal to 22·63 horse-power during one minute, and one calorie per second equals 1·4054 horse-power.

Heats of Combustion.—Exhaustive experiments have been made on numerous elements and chemical compounds to determine the amount of heat generated during the processes of combustion, chemical combination, absorption, and solution. Most of these results are contained in M. M. P. Muir's 'The Elements of Thermal Chemistry,' 1885.

In this book the kilogram and the degree centigrade are employed, and in order to reduce the values to English measure they have to be multiplied by $\frac{9}{5}$, or divided by $536\frac{2}{3}$, to convert them respectively into thermal or into evaporative units.

For convenience in calculating the heats of complicated chemical processes the values in that and similar books are not stated per unit of weight of each element or substance, but per atomic weight. Thus the value for pure carbon is given for 12 kils. and not for 1 kil. of carbon, because the atomic weight of this element is 12.

The following table contains a few of the most important determinations. See also W. Ostwald, 1887.

Table of Heats of Combustion of Elements and Compounds.

Element	Formula	Oxygen required	Atomic Weight	Heats of Combustion: Per Atomic Weight: Thermal Units	Per Atomic Weight: Evaporative Units	Per Pound: Thermal Units	Per Pound: Evaporative Units
Diamond	C	O_2	12	168,000	174·0	14,000	14·50
Native graphite	C	O_2	12	168,500	174·5	14,040	14·53
Amorphous carbon	C	O_2	12	174,500	180·8	14,540	15·07
Carbon imperfectly burnt	C	O	12	52,100	53·9	4,340	4·5
Amorphous silicon	Si	O_2	28	395,000	409·0	14,100	14·6
Sulphur (melted)	S	O_2	32	130,200	135·0	4,070	4·2
Phosphorus (yellow)	P_2	O_5	31	666,000	689·0	21,500	22·3
Hydrogen (product water)	H_2	O	2	123,000	127·4	61,500	63·7
Hydrogen (product steam)	H_2	O	2	105,720	109·4	52,860	54·7
Iron	Fe_3	O_4	177	477,000	493·0	2,751	2·84
Carbonic oxide	CO	O	28	122,400	126·9	4,375	4·5

Partial Combustion.—The heat generated by the combustion of 28 lbs. of carbonic oxide ($CO = 12\ C + 16\ O$) amounts to 126·9 English evaporative units. If these are added to 53·9, which are generated by imperfectly burning 12 lbs. of carbon, 180·8 is obtained, which is exactly equal to the heat evolved by burning these 12 lbs. of carbon to carbonic acid in one operation. This shows that, as regards the final result, it does not matter whether carbon is burnt in one or in two stages, or, in other words, whether it is completely burnt at once, or is first converted into carbonic oxide gas and then burnt. Another feature of interest is that the partial burning of 12 carbon, to form 28 carbonic oxide, only produces 45 % of the heat which will be evolved when 28 carbonic oxide are burnt to carbonic acid. The remaining 55 % might be looked upon as the latent heat of evaporation of 12 lbs. of carbon. This would imply that it requires four and a half times as much heat to evaporate carbon as to evaporate water. But this simple law is not supported by other experiments.

From a study of the heats of combustion of various chemical compounds it may be inferred that the calorific value of any fuel cannot be determined accurately by calculations based on chemical analysis alone.

Heating Power of Fuels.—There are two methods of determining the heating power of fuels: either by measuring the amount of heat generated during their combustion, or by chemically analysing them, and summing up the known amounts of heat which each element would generate if burnt separately. The serious difficulties which at one time beset the first of these methods have been successfully overcome by the use of the so-called

Bomb Calorimeter. (See Berthelot and Vieille, 'An. Ch. et Ph.,' 1887, vi. vol. x. p. 433; I. Mahler, 'I. and S. I.,' 1892, p. 183.)—The fuel is powdered and placed in a small platinum crucible inside a strong iron bomb. Moist oxygen, under a pressure of 350 lbs., is then admitted, and the whole is placed in a calorimeter, and allowed to cool till the water temperature has grown steady. The fuel is ignited by electricity, and is consumed rapidly and completely. The rise of temperature of the calorimeter, amounting to about 5° F., is then the measure of the heat evolved.

The combustion being a perfect one, very few corrections are necessary. Excepting those for ashes and moisture, they do not exceed 1 %. All uncertainty about radiation and the trouble due to the friction of the mercury in the thermometer, have been overcome by the arrangement adopted by C. J. Wilson, who adjusts the temperature of the surrounding envelope so as to agree with the changes of the calorimeter water.

Formerly the only reliable calorific measurements had to be carried out in the elaborate instrument designed by MM. Faber and Silbermann, in which the fuel was consumed in an atmosphere of oxygen under normal pressure.

The escaping gases are led through a long cooling tube, and their temperature measured. They are then collected, and analysis carried out, and, what is of great importance, accurate corrections are made for incomplete combustion.

Thomson Calorimeter. — Fig. 102 shows a simple, and, for comparative purposes, a reliable instrument. The broken (not powdered) fuel is weighed and placed in a small platinum crucible, F, which is supported in a fork by the lower lid of an open-ended bottle, C. This lid is pierced by a few tubes, D, having very small holes at their ends. A tube, O, of fire-proof material reaches through the upper cork down to the fuel. Its one branch is connected by means of an india-rubber tube to an oxygen flask, and its vertical branch is closed by a nipple or stopper. T is a very sensitive thermometer.

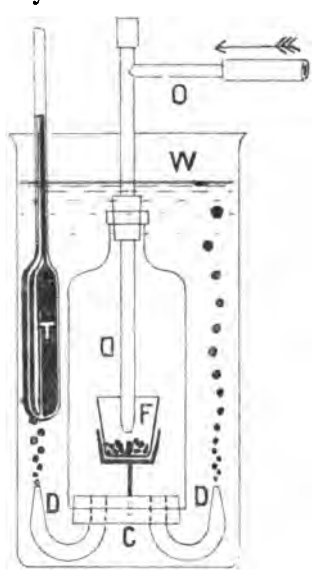


FIG. 102.

When ready the bottle C is lowered into the calorimeter W, containing a weighed quantity of water, and is allowed to remain there till the temperature is steady. Care has to be taken not to allow any

escape of air, otherwise water will enter through the tubes D and flood the fuel. When ready, a small piece of glowing tinder is dropped down the tube O, which is at once closed again, and the blast of oxygen turned on. The products of combustion escape through the tubes D, parting with most of the heat which they contain to the surrounding water. After five or ten minutes the combustion is completed; the water is allowed to enter the bottle, cooling the interior, and is then expelled, and the generated heat can now be estimated by measuring the total rise of temperature of the water, which amounts to from 3° to 6° F. To obtain accurate results various precautions must be taken, of which some of the more important are: reduction of losses by radiation; the use of moist oxygen of the same temperature as the calorimeter, so that none of the calorimeter water is evaporated. This water may be slightly acidulated, so that it will absorb no carbonic acid. The correction for complete absorption in water would be a subtraction of ·25 evaporative unit for each pound of gas absorbed, or ·92 for each pound of carbon burnt. If caustic soda were added to the water, these values would be ·85 and 3·12 respectively. The amount of hydrogen and moisture in the coal, which is all converted into water, not steam, must be allowed for. A further addition of from 7 to 10 % seems necessary for unavoidable losses.

Coal Analysis.—The calculation of the calorific value from the chemical analysis is necessarily somewhat incorrect, for although the heats of combustion of the various elements have been accurately determined, practically nothing is known about the amount of heat required to split up the chemical compounds which constitute coal. It must also be remembered that the amount of oxygen in the coal cannot be determined analytically, but only by difference, and is therefore seriously affected by the chemical composition of the mineral matter in the coal, which is entirely altered, and its weight increased, while being converted into ashes.

Ashes.—It is even possible that these have a direct influence on the heating value of a fuel.

That iron and sulphur form one of the compounds is demonstrated by the presence of pyrites in coal; yet few formulæ take account of this mineral. All the other elements—viz. silicon, calcium, aluminium—doubtless change their chemical condition as the carbon leaves them, and give off or absorb heat as they are converted into ashes or clinker. The heat which they either evolve or absorb depends on whether oxygen is in excess or not, and on the temperature at which combustion takes place. W. Thomson ('Soc. Ch. Ind.,' 1889, vol. viii. p. 525) and Scheurer-Kestner ('An. Ch. et Ph.,' 1886, vi. vol. viii. p. 271) have made numerous determinations, but the results stand in no relation whatsoever to the chemical composition, that of the ashes not being given. A long list of the mineral constituents of coal from various parts of the world will be found in Dr. F. Muck's book, 1891.

Calorific Formulæ.—The following are the formulæ which have been compiled by various authorities for the purpose of enabling the known chemical compositions of a fuel to be used for estimating its heating power:—

C. J. Wilson . .
$$\begin{cases} \dfrac{14540\ C + 52200\ H - 966\ m}{100} = H^1. \\ \dfrac{15{\cdot}07\ C + 54{\cdot}00\ H - 1\ m}{100} = E. \end{cases}$$

Author not known
$$\begin{cases} \dfrac{14540\ C + 62200\ H + 3960\ S}{100} = H^1. \\ \dfrac{15{\cdot}07\ C + 64{\cdot}40\ H + 4{\cdot}10\ S}{100} = E. \end{cases}$$

Scheurer-Kestner
$$\begin{cases} \dfrac{14540\ C + 62200\ H}{100} = H^1. \\ \dfrac{15{\cdot}07 + 64{\cdot}40\ H}{100} = E. \end{cases}$$

Dr. O. Gmelius .
$$\begin{cases} (100 - m - A)\ 144 - K\ m = H^1. \\ (110 - m - A)\ {\cdot}150 - K^1\ m = E. \end{cases}$$

K and K^1 are constants. $K = -7$ when $m \leqq 3\ \%$; $K = +11$ when $m = 3$ to $4\frac{1}{2}\ \%$.

$K^1 = -\frac{7{\cdot}5}{1000}$ when $m \leqq 3\ \%$; $K^1 = +\frac{11}{1000}$ when $m = 3$ to $4\frac{1}{2}\ \%$.

Author not known
$$\begin{cases} \dfrac{14540\ C + 62200\,(H + \frac{m}{9})}{100} = H^1. \\ \dfrac{15{\cdot}07\ C + 64{\cdot}40\ (H + \frac{m}{9})}{100} = E. \end{cases}$$

Dulong, also Grove and Thorpe
$$\begin{cases} \dfrac{14540\ C_1 + 53400\ H - 1148\,(m + m_1)}{100} = H^1. \\ \dfrac{15{\cdot}07\ C + 55{\cdot}30\ H - 1{\cdot}190\ (m + m_1)}{100} = E. \end{cases}$$

m_1 is the moisture generated during combustion, but after drying.

Schwackhöfer . .
$$\begin{cases} \dfrac{14540\ C + 53400\ (H - \frac{O}{8}) + 3600\ S - 1148 m}{100} = H^1. \\ \dfrac{15{\cdot}07\ C + 55{\cdot}30\ (H - \frac{O}{8}) + 3600\ S - 1{\cdot}19\ m}{100} = E. \end{cases}$$

Carnut . . . .
$$\begin{cases} \dfrac{14540\ C_1 + 20200\ C_2 + 62200\ H}{100} = H^1. \\ \dfrac{15{\cdot}07\ C_1 + 20{\cdot}90\ C_2 + 62200\ H}{100} = E. \end{cases}$$

Here C_1 is the percentage of carbon remaining after coking, and C_2 is the carbon which evaporates as hydrocarbon

Welter . . . .
$$\begin{cases} \dfrac{48400\ (\frac{C}{3} + H)}{100} = H^1. \\ \dfrac{50{\cdot}20\ (\frac{C}{3} + K)}{100} = E. \end{cases}$$

In these formulæ C, H, O, and S stand respectively for the percentages of carbon, hydrogen, oxygen, and sulphur contained in the fuel, while A and m are the percentages of ash and moisture. H^1 and E are the calorific and evaporative values of the fuel.

As an example, apply C. J. Wilson's rule to the case of the block fuel used on board the 'Ville de Douvres' ('M.E.,' 1892, p. 142).

$$
\begin{array}{lrcr}
84{\cdot}65 \text{ per cent. carbon} & \times\ 15{\cdot}07 & = & 1{,}275 \\
3{\cdot}98 \text{ ,, ,, hydrogen} & \times\ 54{\cdot}00 & = & 215 \\
 & & & 1{,}490 \\
2{\cdot}41 \text{ ,, ,, moisture} & \times\ 1 & = & 2 \\
 & & & 1{,}488 \div 100 = 14{\cdot}88
\end{array}
$$

Numerous experiments made by several of the above authors and independent experimenters lead to the conclusion that none of the formulæ can be relied upon absolutely.

The following analyses and calorific values, as determined by the bomb calorimeter, of the fuel used on the trials of the Research Committee and a few others have been kindly supplied by Mr. C. J. Wilson :—

Vessel's Name and Fuel	Composition of Fuel in %					Calorific Value	Evaporative Value
	C	H	O+S+N	Ash	Moist.		
'Colchester' { Yorkshire / Nottinghamshire }	71·89	5·42	14·36	4·08	4·25	13,176	13·63
'Tartar' { Penrikyber / Welsh coal }	87·98	4·22	3·31	3·42	1·07	14,765	15·28
A sample of Welsh coal	81·76	3·81	2·46	9·32	2·65	13,802	14·30
A sample of coal	86·02	4·22	4·48	3·94	1·34	14,710	15·25
A sample of coke	76·94	0	0	14·23	8·83	11,032	11·40
A sample of coke	71·17	0	0	19·46	9·37	10,200	10·58

The following table contains summaries of some analyses and calorific determinations (W. Thomson, 'Soc. Ch. Ind.,' 1889, vol. viii. p. 525) :—

Name of Fuel	Composition of Fuel in %							Experimental	
	C	H	O	S	N	Moist.	Ash	Calorific Value	Evaporative Value
Nixon Navigation Glam.	88·03	4·11	1·98	·66	·96	1·02	3·22	15,010	15·55
Tyldesley, Theperley & Co.	68·13	4·78	4·86	1·39	1·22	4·72	14·90	11,610	12·20
Tyldesley Coal Co. . .	74·46	5·10	8·25	·49	1·53	6·07	4·09	12,730	13·18
Upper Drumgray seam { Bottom .	75·48	4·98	7·86	·75	1·59	2·78	6·55	13,300	13·77
Upper Drumgray seam { Mid rib .	72·13	4·67	6·56	·54	1·25	2·26	12·58	12,520	12·96
Upper Drumgray seam { Top seam .	75·05	5·12	9·39	·86	1·76	3·53	4·29	13,560	14·05
Bickshaw Main . .	78·93	4·90	7·24	1·04	1·56	4·36	1·96	13,430	13·91
Pemberton 5-ft. . .	72·41	5·16	8·84	·93	1·41	6·70	4·55	13,040	13·50
Cramhawke . . .	69·77	4·82	12·44	1·17	1·33	7·15	3·31	13,420	13·89
Wigan 4-ft. . . .	76·49	4·96	8·46	1·07	1·44	4·84	2·75	13,590	14·07
Bickershaw 7-ft. . .	73·91	4·86	11·32	·68	1·67	6·60	·96	13,350	13·82
Pendleton 4-ft. . .	79·76	4·89	7·51	·59	1·43	3·90	1·91	13,920	14·42

These calorific values were obtained in a Thomson calorimeter and corrected, while the following, made by Scheurer-Kestner ('An. Ch. Ph.,' 1886, vi. vol. viii. p. 271), were made in the more elaborate calorimeter of Faber and Silbermann.

Name of Fuel	Composition of Pure Fuel in %					Pure Coal	
	C	H	O+S+N	Ash to be added	Coke produced	Calorific Value	Evaporative Value
Bwlf	91·08	3·83	5·09	3·32	82·1	15,800	16·4
Powels	92·49	4·04	3·47	3·72	87·2	16,060	16·6
Altendorf	89·92	4·11	6·0	10 to 12	83·9	16,400	17·0
Ronchamp	89·09	5·09	5·82	—	78·8	16,420	17·0

Nixon's Coal.—Scheurer-Kestner ('Soc. I. Mul.,' 1888, vol. lviii. p. 313):—

C	H	O	S	N	Ash.	Calorific Value	Evaporative Value
85·73	4·17	3·97	·66	·46	5·01	15,950	16·5

Prof. H. Fritz ('Dingler's J.,' 1876, vol. ccxix. p. 185) gives the following list:—

Name of Fuel	Calorific Value	Evaporative Value	Name of Fuel	Calorific Value	Evaporative Value
Hydrogen	62,100	64·3	Coke (15 % cinder)	10,430	10·8
Coal gas	39,600	41·0	Alcohol (pure)	12,520	12·9
Petroleum	18,200	18·8	Peat coal	10,430	10·8
Olive oil	17,630	18·2	Lignite	9,000	9·3
Wax	15,660	16·2	Peat (dry)	8,640	8·9
Tallow	14,930	15·4	Peat (20 % moisture)	6,480	6·7
Anthracite	14,590	15·1	Red coal	7,160	7·4
Carbon	14,540	15·0	Dry wood	6,480	6·7
Coal (mean)	13,500	14·0	Wood (20 % moisture)	5,040	5·2
Charcoal	12,600	13·0	Straw	3,360	3·5
Coke (clean)	12,600	13·0	Tan	5,580	5·7

Berthier's Method is based on the fallacious idea that the amount of heat generated by burning a fuel is proportional to the oxygen consumed, but being a fairly simple one, it may occasionally be of use. It is carried out as follows:—Accurately weigh, say, 25 grains of fuel, and mix them intimately with 2 oz. of pure litharge, and pour them into a crucible (fig. 103), B and F. They are then covered with 1½ oz. of pure litharge, L, and over all is placed a layer of ¼ inch of broken glass, G. The crucible is then heated to redness, care being taken that the top grows hot first, so as to melt the glass, and that it is well protected from smoke. After about three-quarters of an hour's heating, it is allowed to cool, and is then broken up. It will be found to contain a button of lead, B. This is hammered and washed to remove all impurities. It will weigh from 700 to 800 grains. Whatever the result, divide it by the weight of fuel used, and multiply by the constant C_1, so as to obtain its calorific value, or by C_2 to obtain its evaporative value.

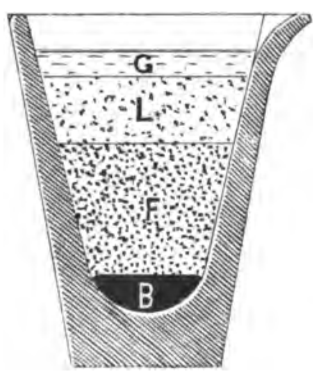


FIG. 103.

Nature of Fuel	Constants	
	C_1	C_2
Anthracite, charcoal, &c.	543	·563
Welsh and similar coal, yielding, say, 90 % coke .	518	·537
Bituminous coal, yielding 65–85 % coke . .	500	·517

See 'An. Ind.,' 1883, vol. i. p. 38.

The reason why the coefficients vary is because the hydrogen in the fuel produces four times as much heat as an equal weight of carbon, but only reduces three times as much lead.

Another way of roughly estimating the heating value of a fuel is to ascertain the amount of moisture, ashes, and volatile compounds which it contains, and to credit every pound of the latter as being able to evaporate twenty times its weight of water.

The following table is a summary of C. Grove and W. Thorpe's (1889, p. 54) analysis of 111 samples of fuel; it contains only the highest and lowest values, and clearly shows how much their compositions vary.

Analysis of Coals.

Coal Field	No. of Samples	—	C	H	O	S	N	Ash	Density	Coke
Welsh	37	Max.	91·4	5·8	17·9	5·1	2·2	14·7	1·38	92·9
		Min.	71·1	3·5	·4	·1	0	1·2	1·25	55·4
Newcastle	18	Max.	86·8	6·7	10·3	2·8	1·8	9·1	1·31	72·3
		Min.	78·0	4·5	2·4	·1	·7	·2	1·25	35·6
Derbyshire	7	Max.	81·9	5·6	12·4	1·3	2·1	4·6	1·32	62·5
		Min.	78·0	4·8	8·6	·7	·8	1·2	1·27	52·8
Lancashire	28	Max.	83·9	6·2	20·0	3·0	2·2	14·3	1·35	66·1
		Min.	72·9	4·8	4·9	·5	·5	1·1	1·23	51·1
Scotch	7	Max.	88·5	6·5	15·5	1·6	2·0	10·7	1·32	59·1
		Min.	74·5	4·8	·9	·4	0	1·2	1·23	49·3
Van Diemen's Land . .	8	Max.	70·4	4·2	9·1	1·3	1·6	30·4	—	—
		Min.	57·2	2·9	1·0	·7	·9	14·4	—	—
Chili	6	Max.	78·3	6·4	17·3	6·1	1·1	36·9	—	—
		Min.	39·0	4·0	8·4	·9	·5	5·7	—	—

The object of determining the heating power of a fuel is undoubtedly to obtain a correct measure of its commercial value; but heating power is not everything, and there are other qualities, such as liability to smoke, to flame, to cake, to ignite easily and even spontaneously, to break up into dust when exposed, and various others.

Burning Qualities.—Amongst these one of great importance is as to whether a coal will ignite easily or not. Thus Cannel coal can be lit by applying a match, while anthracite can only be kept alight under carefully regulated conditions of draught, &c. The success of explosives depends on a good knowledge of this subject; thus we find that sulphur, which ignites at a low temperature, has to be added to gunpowder in larger or smaller quantities, according as to whether it has to explode quickly or slowly. More recently it has been discovered that the temperature at which wood is converted into charcoal materially affects the temperature at which it ignites, and that where the process is carried out gently and over a long period, as is the case with wooden beams in houses placed close to flues, ignition ultimately takes

place spontaneously. Gun cotton ignites so readily that it could not be used for ammunition until it was discovered that an admixture of camphor or nitro-glycerine raised this temperature; chloride of tin still further reduces its liability to ignite. Some curious phenomena have been noted. Thus phosphorus, which ignites at a temperature of 140° F. in an ordinary atmosphere, must be raised to a much higher temperature if placed in absolutely pure oxygen. A mixture of hydrogen and chlorine gas ignites at a very low temperature if illuminated by actinic rays, but no amount of heating, short of redness, will effect their chemical combination in the dark.

Table of Temperatures of Ignition.

Substance.	Igniting Temperatures. °F.
Phosphoretted hydrogen	116
Phosphorus	140
Sulphur	470
Carbon disulphide	300 or 440
Dried peat	435
Lignite dust	300
Anthracite dust	570
Coal	600
Charcoal made at 550° to 700° F.	680
„ „ 2,200° to 2,300° F.	1,100 or 1,400
Cokes	Red heat
Anthracite	„ 750
Carbonic oxide	„ 1,211
Hydrogen	1,030 or 1,290
Fire damp	1,200

With hydrocarbons the igniting temperature depends on the chemical composition; thus benzine ignites at about 500° F., while some of the dense oils will quench red-hot iron. Generally speaking hydrocarbons prevent both quick and spontaneous combustion.

More precise information on igniting temperatures would greatly assist the understanding of fuel combustion, but much may be learnt about the behaviour of various coals by consulting the following papers:—'Memoirs of the Geological Survey of Great Britain' (London, 1848, vol. ii. p. 539); Report of Sir H. De la Bêche and Dr. Lyon Playfair.

Coal Trials.—Parliamentary Papers.

Date	No.	Index Vol.	Index No.	Remarks
1857–8	375	39	43	
1859	116	25	209	Welsh and Newcastle
1860	485	42	267	
1862	204	34	113	
„	337	„	11	
„	364	„	121	Australian
1863	79	35	151	
„	159	„	141	„
1864	(80, 80–I.)	37	187, 211	Welsh and Newcastle
„	375	„	213	„ „
1865	365	35 II.	79	

Coal Trials.—Parliamentary Papers—cont.

Date	No.	Index Vol.	Index No.	Remarks
1866	440	46	35	
1867	561	44	339	
,,	563	,,	491	Lancashire and Cheshire
1868–9	212	38	443	
,,	270	,,	447	,, ,,
1870–71	121	40	463	Newcastle
1872	165	39	349	
1873	—	42	537	Indian troopships
1876	280	47	713	
1877	397	52	515	
1878	396	49	541	
,,	233	47	495	
1878–9	380	45	493	

Spontaneous Combustion of Coals.— Parliamentary Papers.

Date	No.	Index Vol.	Index No.
1876	C. 1586–1	41	1
1878	366	47	33
1878–9	373	64	47
1880	C. 1586–1	41	1
1886	366	59	45

Influence of Air Pressure on Combustion.— Frankland's experiments (1877, p. 863) show that, contrary to the accepted views, increased pressure (not draught) does not accelerate combustion, but that it increases the luminosity and smokiness of flames. At twenty atmospheres air-pressure the hydrogen flame gives a perfectly continuous spectrum, while the carbonic oxide flame is equally luminous at fourteen atmospheres. An ordinary candle gives hardly any light when burnt on the summit of Mont Blanc, but smokes under pressure.

Temperatures of Flames.—When no heat is lost by radiation or convection the entire quantity generated during combustion must have been used to warm the products, raising their temperature sufficiently to make them glow, and the result is a flame. Its temperature is calculated as follows :—If n pounds of air are supplied for every pound of carbon burnt, then the weight of the products will be $n+1$, the heat evolved will be 14,540 cals. per pound of fuel, and if σ is the specific heat of the gases, the temperature of the flame $T = \frac{14540}{\sigma.(n+1)}$.

The Specific Heat of a Gas can only be accurately estimated if the specific heats of individual substances are known. The following small list contains all the necessary data:—

Table of Specific Heats at Constant Pressure.

ELEMENTS.

Carbon (charcoal) .	·1935	Oxygen	·2175
Hydrogen	3·4090	Nitrogen	·2438

COMPOUNDS.

Air	·2375	Water	1·0000
Carbonic oxide (CO) .	·2450	Marsh gas (CH_4) . .	·5929
" acid (CO_2) .	·2163	Sulphurous acid (SO_2) .	·1540
Steam (H_2O) . .	·4805		

There is some uncertainty as to the specific heat of steam, but as it does not enter very largely into the composition of burnt products, the value ·4805 will be accepted. J. Macfarlane Gray ('Phil. Mag.,' 1882, vol. xiii. p. 337) estimates it at ·379.

Air being composed of 20·8 volumes of oxygen and 79·2 volumes of nitrogen, or by weight 23 oxygen and 77 nitrogen, its specific heat is

$$\frac{23 \times \cdot 2157 + 77 \times \cdot 2434}{100} = \cdot 2375, \text{ as stated in the table.}$$

The quantity of oxygen required to consume 1 lb. of carbon is $\frac{32}{12}$=2·667, and this carries with it 8·92 lbs. of nitrogen.

The calculation is therefore as follows :—

$$\begin{array}{lll} 1\cdot000 & \text{carbon} & \\ 2\cdot667 & \text{oxygen} & \\ \hline 3\cdot667 & \text{carbonic acid} & \times \cdot2163 = \ \ \cdot793 \\ 8\cdot920 & \text{nitrogen} & \times \cdot2438 = 2\cdot171 \\ \hline 12\cdot587 & \text{products} & \times \cdot2355 = 2\cdot964 \end{array}$$

So that the specific heat of this mixed gas is ·2355. But the amount of air supplied, viz. 11·6 times as much as the carbon, is only just sufficient for perfect combustion ; in practice it is above 15 and sometimes even over 30. By adding 7·413 lbs. of air to the above the products of combustion are increased to 20 lbs., and the specific heat would be raised to ·2360.

Coal contains both hydrogen and some moisture, and, as the specific heat of steam is about twice as great as that of dry air, the average value is somewhat higher than the above. However for practical purposes the value ·237 is sufficiently accurate.

Flame Temperature.—It is now possible to determine the temperature T of the flame with the help of the previous formula. Let $n+1=20$, then T=3,080° F. above t, the mean temperature of the air and fuel. Roughly speaking, the furnace temperature multiplied into the ratio of products to coal ($=n+\cdot 9$) is 61,000° F. ; but only down to a limit of about $n=15$. Below this point the combustion of the coal is not a perfect one, and instead of 14,540 calories only 4,340 are evolved.

Unless much hydrocarbon gas or smoke is produced we have the following three conditions :—

When $n=6\cdot 8$, carbonic oxide and nitrogen are the only products ; the initial temperature is 2,350° F.$+t$.

When n is greater than 6·8 and less than 15, the products consist of carbonic oxide, carbonic acid, nitrogen, and oxygen, and the temperature grows higher till $n=15$ is reached ; then no carbonic oxide is evolved. The initial temperature is 4,130° F.$+t$.

Above this point the temperature grows less and less, as stated above, viz. $\frac{61,000}{n+\cdot 9}$.

When 1 lb. of hydrogen is being burnt, 8 times its weight of oxygen is required, which is accompanied by 26·8 lbs. of nitrogen, making a total of 35·8 lbs. of products of combustion. These consist of 9 lbs. of steam and 26·8 of nitrogen. The specific heat of this mixture is, say, $\frac{9 \times \cdot 480 + 26 \cdot 8 \times \cdot 2434}{35 \cdot 8} = \cdot 303$ (or ·278). The heat evolved being 52,860 cals., the temperature of this flame is 4,873° F. + t.

For such cases in which $n+1$ exceeds, say, 50, divide 247,000 by $(7 \cdot 7 + n)$ and the quotient is the temperature.

The flame temperature of a compound containing carbon and hydrogen would of course depend on some as yet undiscovered law, but a fairly good guess can be made by taking a mean of the values just found. Roughly

$$T = \frac{61,000\ C + 247,000\ H}{(C + H)n + C \cdot 9 + H 7 \cdot 7}.$$

Of course the actual temperatures of flames are decidedly lower than those found by this formula, because loss of heat takes place by radiation during the process of combustion.[1]

The following are a few particulars of temperatures of flames :—

('*Franklin Inst.*,' 1878, *vol. lxxvi. p.* 205.)

Bunsen burner using 1 vol. gas, 2·4 air gave	2,480° F.
" " " 1 vol. gas, 2·4 air, and 1 nitrogen gave	2,150 "
Do. do. and 2 nitrogen gave	1,890 "
Bunsen burner, 1 vol. gas, 2·4 air and CO_2, gave	2,010 "
" " " " $2CO_2$ "	1,434 "
Locatelli lamp	1,690 "
Stearine candle	1,723 "
Petroleum lamp with chimney	1,885 "
" " without "	1,690 "
" " sooty envelope	1,434 "
Alcohol (·912) lamp	2,140 "
" (·822) "	2,150 "

H. Le Chatelier, '*Comp. Rend*,' 1892, *vol. cxiv. p.* 470.

Bessemer process of steel-making	2,984° F.
Siemen's " "	2,876 "
Glass-melting	2,390 "
Electric light	3,272–3,812 "

The Temperature of Dissociation.—When this temperature is reached chemical compounds are split up. For sympathetic ink and cupreous hydrate it is low, for gypsum it is higher, also for carbonate of lime, sulphate of iron, &c. For steam and for carbonic acid it is very high and has not yet been determined. From this it would follow that the higher the initial temperatures of the fuels and air the less heat is evolved, until a point is reached where no combustion takes place. This is far higher than any temperature to be met with even in

[1] Moisture in the atmosphere seriously affects boiler performances.

a steel melting furnace, but its existence must not be lost sight of where heated air or gas is used, for the expected benefit will not be fully realised, nor will the flame temperatures be as high as expected.

Furnace Temperatures.—Hardly any experiments have yet been made to determine furnace temperatures, but since the introduction of H. Le Chatelier's electric pyrometer ('M. E.,' 1891, p. 547, plate 115) and MM. Mesuré and Noël's optical one ('I. and S. I.' 1889, p. 251) at least one of the difficulties has been removed. The first of these instruments is a thermopile, the heated junction of the circuit consisting of wires made of two different alloys of platinum. The optical pyrometer consists of a tube in which a quartz plate is fixed near the centre, and a Nichol's prism at either end. One of these can be turned round its own axis. On looking at a heated object, and slowly turning the prism, it will be noticed that the colour changes from green to red and *vice versâ*. The angle at which this change takes place is noted, and is the measure of the temperature. Care has to be taken not to let any daylight be reflected from the hot object, otherwise the readings will be those of the temperature of the sun. Both these instruments have to be graduated according to some reliable standard.

Air Thermometer.—If properly manipulated there is probably no more accurate measure of temperature than an air thermometer. All the permanent gases expand at the rate of ·3665 between the freezing and the boiling point of water. This is at the rate of ·0020611 per °F. It has been assumed that the **absolute zero** is reached when the volume of air is reduced to nothing. At the above rate this would be found to be 491° below the freezing point of water, or $-459°$ F. Therefore if t is the Fahrenheit temperature of an object, its absolute temperature would be $T=t+459°$. The instrument shown in fig. 104 is based on the above facts. It consists of a small pear-shaped platinum bulb, having a small pin-hole at its apex. It is carefully filled with nitrogen gas, enclosed in fire clay, and placed in the furnace whose temperature is to be measured. When sufficiently heated the whole instrument is plunged into cold or boiling water, and kept there point downwards till it has acquired the temperature of its surroundings. The protecting scale of fire clay has of course fallen off. The bulb is now carefully weighed, first in its present condition, viz. partly filled with water, then when quite full, and then when quite empty. The ratio of the total internal volume of the bulb, which, while in the furnace, was full of hot air, to the volume of air remaining after cooling in the water bath, is exactly the ratio of the absolute furnace temperature to that of the bath. Any other pyrometer can be graduated by placing it alongside the above.

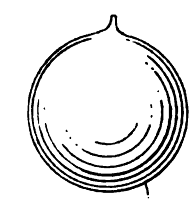


FIG. 104.

Pyrometers.—Descriptions of various pyrometers will be found in the following publications :—

J. Wilson, 'M. E.,' 1852, p. 53, mentions one which consists of a piece of platinum, to be thrown into a calorimeter while hot.

M. Launy, 'Comp. Rend.,' 1869, vol. lxix. p. 347. The pressure exerted by the liberated carbonic acid gâs from heated lime is measured.

T. Carnelly and T. Burton, 'I. and S. I.,' 1884, p. 195. Various pyrometers.

Prof. J. Wiborg, 'I. and S. I.,' 1888, ii. p. 110. An exposed bulb is filled with compressed air.

M. Santignon, 'Génie C.,' 1890, vol. xvi. p. 328. This pyrometer consists of a bent metal tube which is inserted in the heated space, and a steady current of water is made to flow through it.

Messrs. Heish and Folkard, 'Iron Age,' vol. xxxvii. p. 25. Platinum bulb air pyrometer.

Landolt and Börnstein (1883) give a long list of melting temperatures of various alloys and metals. (See also p. 86.)

Temperatures in the Fuel.—It has already been explained that the temperature of a flame is proportional to the percentage of carbonic acid it contains till a point is reached where carbonic oxide is formed. Fig. 105 illustrates, in a rough way, the conditions which may be expected in the interior of a thick fire.[1]

Estimated Ratio of Air to consumed Carbon	Thickness of Fire	Estimated Temperatures in the Interior of a Fire
20 : 1. Air admitted above fuel		2,960° F.
6·8 : 1. Chiefly CO . .		2,410° F.
10 : 1. Partly CO and CO_2		2,980° F.
15 : 1. { Complete combustion . . .		3,870° F.
20 : 1. Do. . A. —		— A. 2,960° F.
43 : 1. Do. . . .		1,550° F.
1 : 0.		60° F.

FIG. 105.

Of course this picture is mere guesswork, but it shows that by measuring the temperatures at various depths, or by analysing the gases at these points, some valuable information might still be obtained as to the process of combustion (compare p. 244).

Prof. A. Ledebeur ('Stahl und Eisen,' 1882, vol. ii. p. 356) made the following interesting experiments, in which the temperatures of the burning charcoal (contained in a tube) were carefully regulated. To a certain extent these experiments modify the above calculations: —

Temperature		Percentage of Carbon burnt		Oxygen used
	Degrees Fahrenheit	CO_2	CO	%
—	660	78·6	21·4	33·0
—	710	72·4	27·6	80·6
Dark red . .	1,150	71·4	28·6	87·9
Cherry red .	1,250	62·6	37·4	80·3
Yellow . .	2,010	1·3	98·7	100·0

[1] In blast furnaces no free oxygen and hardly any carbonic acid are found at a greater distance than two feet from each tuyer (p. 244).

With thin fires it is quite possible that sufficient air for complete combustion passes through the fuel. In this case the conditions above the line A do not exist, and it is not necessary to admit air above the bars; but with thick fires, particularly with a weak draught, the upper layers of fuel effect the reduction of carbonic acid to carbonic oxide, and air must be admitted above. It will also readily suggest itself that if green coal is thrown on a fire which is still evolving much carbonic oxide, the latter will be cooled to such an extent that it cannot ignite when coming in contact with the upper air. Fires should, therefore, be allowed to burn as low as possible before recoaling.

It might seem to be an advantage, even in cases of forced draught, to keep the fires thin, and thereby prevent the formation of combustible gases. This, however, is not the case, for, unless the grate surface were very much increased, there would be a serious loss, due to the blowing away of unburnt coal particles.

Effects of Draught.—In cases of complete combustion the weight of the products is sometimes thirty times the weight of fuel burnt. Now, suppose that the consumption is raised from 20 lbs. per square foot per hour to 40 lbs.; then the velocity of the escaping gases, which amounts to 600 lbs. per hour = 10 cubic ft. per second, would be twice as great, and the power to carry away solid particles would have been increased about fourfold. If, however, the fires are kept thick and all the gases leave the fuel as carbonic oxide, their weight is reduced to 270 lbs., and the temperature being slightly lower, and the bulk proportionately less, the power to lift up and carry away small particles of coal is reduced to about one-fifth. Of course even the thickest fires in a marine boiler will allow free oxygen and much carbonic acid to pass, so that the above favourable condition is hardly approached; but at any rate it is clear that with forced draught fires should be kept thick, restricting the admission of air under the bars, keeping the temperatures high, and giving much fuel contact to the carbonic acid gases, so as to convert them into combustible ones. A very large proportion of air should then be admitted at the doors or bridges. Naturally the horizontal draught will be excessive unless the furnaces are very large.

The air which is admitted for the combustion of the gases should be as hot as possible, and when admitted at the bridges the utmost has probably been done in this direction by having passed it under the fire bars. When admitted at the doors artificial heating is the only means of warming it, but, as all draught over the fire bed has the effect of blowing away particles of coal and cinder, this trouble would only be aggravated if the air temperature and bulk were increased. An additional power to do harm is given to the draught if much air is also driven through the bed of coal, whereby all the smaller particles are carried to the top. This can only be prevented by a thick bed of coal, and this again is only possible with large furnaces.

Forced and Natural Draught.—A few formulæ are necessary for explaining this subject, but they are of the simplest.

$$v^2 = 2.g.h.$$

Here v is the velocity of air, h column of air measured in feet, corresponding to the draught pressure, and $g = 32 \cdot 2$ feet is the acceleration

due to gravity. Instead of an air column or mercury column, it is usual to express draught pressure in inches of water. The respective weights of mercury, water, and air being 13·60, 1, and ·001293, the formula is changed into

$$v^2 = 4{,}140 \text{ H} = 56{,}300 \text{ M},$$

where H = pressure measured in inches of water, and M = pressure measured in inches of mercury. In practice the friction and bends will seriously reduce the velocity, and the coefficients have to be reduced by about 25 %. It is also evident that with the same draught pressure an incandescent, and therefore lighter, gas is more easily moved than a cold one. If T denotes its absolute temperature, while 491° is the absolute temperature at 32° F., then $v^2 = 6 \,.\, \text{H} \,.\, \text{T}$ nearly.

It is as well to mention that a more general formula, but without the above correction, would be

$$v^2 = 121{\cdot}5 \frac{\text{H} \,.\, \text{T}}{m}.$$

Here m is the molecular weight of a gas. This is 14·37 for air, about 16 for oxygen, 14 for nitrogen, 22 for carbonic acid, and 9 for steam.

In order to find V, the volume of air discharged per second, the sectional area A, in square feet, of the orifice or channel has to be multiplied by the velocity v.

$$\text{V} = v \,.\, \text{A} = \text{A} \,.\, \sqrt{6 \,.\, \text{H} \,.\, \text{T}}$$

The weight Q is found by multiplying this volume into the weight of a cubic foot of air, and introducing the correction for temperature.

$$\text{Q} = \text{A} \,.\, 97{\cdot}3 \sqrt{\frac{\text{H}}{\text{T}}}, \text{ or roughly } 100 \,.\, \text{A} \sqrt{\frac{\text{H}}{\text{T}}},$$

from which it follows that the pressure H required for delivering Q lbs. of hot air per second through the section A is

$$\text{H} = \frac{\text{Q}^2}{\text{A}^2} \frac{\text{T}}{10{,}000}, \text{ measured in inches of water.}$$

Resistances in Furnaces.—This formula has been used in estimating some of the values in the following table, in which an attempt is made to illustrate what takes place in two furnaces, whose diameters are respectively 30 ins. and 45 ins., with 6-ft. grates. The sectional areas of the ashpits and over the fuel would be 1·6 and 3·7 sq. ft. 20 lbs. per square foot is to be the coal consumption under natural and 40 lbs. under forced draught. The products of combustion will weigh about 24 times as much as the fuel, and it is assumed that two-thirds of the air passes through the fuel, and the other third through the door. If the air is admitted at a temperature of 32° F., it will be reasonable to assume that the temperatures in the fuel and in the flame are about 2,500° F., and that as they pass into the tubes this has been reduced to 1,000° F., while on entering the funnel it has fallen to 600° F.

Estimated Temperatures, Volumes, Velocities, and Resistances of Gases in a Pair of Boiler Furnaces.

Particulars		Draught			
		Ordinary		Forced	
Furnace diameters	ins.	30	45	30	45
Ashpit = $\frac{1}{3}$ sectional area of furnace	sq. ft.	1·6	3·7	1·6	3·7
Sum of sectional area of tubes	,,	4	6	2	3
Grate surface (6-ft. bars)	,,	15	22½	15	22½
Funnel area	,,	3	4½	3	4½
Coal consumption per hour	lbs.	300	450	600	900
Weight of products per second	,,	2	3	4	6
Volume of $\frac{2}{3}$ air (32° F.) through ashpit	cub. ft.	15·8	23·7	31·6	47·5
Mean velocity in ashpit	ft. per sec.	9·9	6·4	13·1	8·5
Resistance	ins.	·076	·032	·31	·13
Volume of $\frac{2}{3}$ products in interior of fuel (2,500° F.)	cub. ft.	96	144	192	289
Upward velocity in fuel	ft. per sec.	6·4	6·4	12·8	12·8
Resistance (*see footnote* [1])	ins.	·0053	·0053	·023	·023
Volume of all products above fuel (2,500° F.)	cub. ft.	144	216	289	433
Horizontal volocity over fuel	ft. per sec.	90	58	119	77
Resistance	ins.	·47	·20	1·63	·79
Volume of products at 1,000° F.	cub. ft.	72	108	144	216
Velocity on entering small tubes	ft. per sec.	18	18	72	72
Resistance	ins.	·037	·037	·60	·60
Volume of products at 600° F.	cub. ft.	53	79	105	158
Velocity in funnel	ft. per sec.	17½	17½	35	35
Resistance	ins.	·047	·047	·19	·19

Doubtful though some of the above results are, they will at any rate serve to make comparisons, and then there can be no question that the large furnace has the advantage.

Closed Ashpits.—When air is blown under the bars it is, perhaps, an advantage to keep the air spaces between them very small, thereby increasing this resistance to such an extent that all the others sink into insignificance. This would ensure a fairly constant supply of air, no matter what condition the fires are in. With a fierce combustion and small furnaces it may then be found necessary to keep the furnace doors open, so as to admit sufficient air above the bars.

Minimum Air Admission above the Grate.—This question is one of the most difficult that presents itself in the working of boilers, and has engaged the attention of engineers for the last fifty years, with no

[1] The resistances in this line have been calculated as if neither bars nor coals existed there; but assuming that the interstices of the fuel are reduced in this channel to 20 % of the above, the upward velocity of the gases would be increased fivefold, and their resistances twenty-five-fold, raising them to, say, $\frac{1}{8}$ in. and $\frac{5}{8}$ in. for natural and forced draught respectively.

better result than the evolution of an idea that air must be admitted above the bars, but as to how much and by what means has never yet been settled. The question of how little can only be determined by chemistry corrected by experience. In the trials of the 'Colchester' ('M. E.,' 1890, p. 206) there are three readings in which from 1 to 2 % of carbonic oxide was found in the waste products. The amount of unconsumed oxygen ranged from 5·4 to 6·6 %, corresponding to from 17 to 18½ lbs. of waste products per lb. of coal. On the other hand, there is one sample of gas in this trial where the oxygen had been reduced to 6·43 %, without evolution of carbonic oxide, and in the case of the 'Ville de Douvres' ('M. E.,' 1892, p. 8) there is one with 5·23 % oxygen. This may be explained by only one of the furnaces of the 'Colchester' working badly, and there, perhaps, the oxygen may have been entirely consumed. To be on the safe side, 5 % of free oxygen, corresponding to 16½ lbs. of waste products per lb. of coal, may be looked upon as the lowest limit. Under such conditions the weight of the escaping gases will be a minimum, and, as their temperature is high, they will more readily part with their heat, and weighing less than when more air is admitted, they move more slowly in the tubes, and have more time to cool down. (See chapter on 'Heat Transmission.')

Maximum Air Admission above the Grate.—It would, therefore, appear that as little air should be admitted as is compatible with perfect combustion; but this is not borne out in practice, as will be seen from the following table, which contains some of the results of trials by the Research Committee of the Institute of Mechanical Engineers, and previously referred to:—

Name of Vessel	Ratio of Waste Products to Fuel	Heat carried away by Waste Products per Cent.	Weight of Fuel consumed divided by Total Heating Surface
'Colchester' . .	18·5	28·0	·987
'Ville de Douvres' .	17·9	26·8	·101
'Fusi Yama' . .	22·8	23·5	·437
'Tartar' . . .	31·5	22·1	·367
'Iona' . . .	25·5	16·2	·298

It would, in fact, appear as if the order of things had been reversed, and that the more air was admitted the more economical was the boiler; but this view leaves out of account the ratio of fuel burnt per hour to heating surface, which value will be found in the last column, and which seems to exert a far greater influence than the other condition. The only way of settling this question would be to carry out progressive trials with the same boiler. By a comparison of the results obtained on the 'Fusi Yama' and on the 'Tartar,' where the waste products were respectively 23 and 31½ times as heavy as the fuel burnt, and the performances of the boilers fairly equal, it will be found that, if anything, less heat was carried away when a large excess (about twice as much as the necessary quantity) of air had been admitted to the fires, showing that little, if any, harm is done by such proceedings.

These remarks apply, of course, only to the economical side of the question, for it is well known that by opening the doors two influences are at work to reduce the boiler performance—firstly, the uptake

temperature, and with it the draught is reduced, and, on account of the relatively smaller resistance encountered by the air above the bars, less air passes through the fuel, and its consumption is therefore seriously checked. With forced draught the available pressure can be increased to practically an unlimited extent, and it ought not to be difficult to regulate its flow.

Velocity of Waste Products.—If anemometers are placed in the ashpits it will be found that their readings cannot be depended upon, as these vary considerably, according to the positions of the instruments. If placed in the uptakes they will soon get damaged by the heat and dirt, and would be difficult to read. Besides, it has not yet been ascertained at what point of the section of a funnel the velocity of the gases is just equal to the mean. These difficulties can, to a certain extent, be overcome by making the arms of the anemometer equal to one quarter of the diameter of the funnel (see fig. 106) and using four vanes, V, set at an angle of 45° to the axis. It will then be found that the time occupied by each vane in sweeping through part of any annular zone of air in the funnel is nearly proportional to the area of that zone, and therefore the readings will be very near the average velocity. A spring or weight should be attached to the clockwork, C, of the instrument, of just sufficient force to overcome the initial friction. The most accurate results would doubtless be obtained if the anemometer were made of the same diameter as the funnel, and its blades shaped like those of the old-fashioned screw propellers, their depth and pitch being kept constant and their widths increasing towards the extremities. In this case it can also be shown that the mean velocity of the gas for all the zones would be recorded.

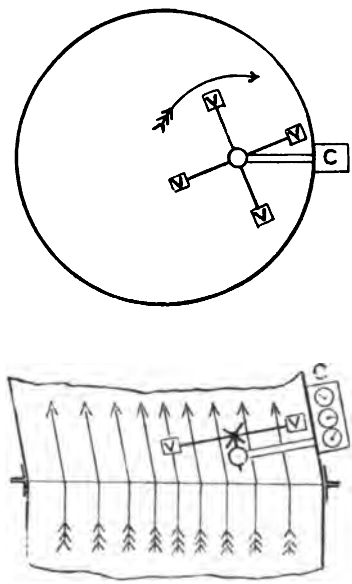


FIG. 106.

The Temperature of the Gases has also to be taken, and, if possible, two or more thermometers should be used, so as to obtain average results, and the necessary data are now available for calculating the weight of the waste products. Let V cubic feet be the measured volume which has passed away in one hour; then if T is its absolute temperature, while 491° is the absolute temperature at 0° F., the weight of air per hour is $Q = \frac{\cdot 0808 \, . \, 491 \, . \, V}{T} = 39{\cdot}7 \frac{V}{T}$. Here ·0808 is the weight of 1 cubic foot of dry air.

If $T = 918°$ and $V = 1{,}000{,}000$, then $Q = 43{,}200$ lbs.

The necessary allowance should of course be made for the height of the barometer, which is here assumed to be 30 inches, and for the specific weight of the waste gases.

—	Specific Gravity	Weight per Cubic Foot. Lbs.	Molecular Weight
Air (dry)	·001293	·0808	14·37
Oxygen	·001430	·0894	15·96
Nitrogen	·001256	·0785	14·02
Hydrogen	·0000896	·0056	1·00
Steam	·000806	·0504	8·98
Carbonic oxide	·001234	·0771	13·96
„ acid	·001977	·1235	21·94
Sulphurous acid	·002740	·1720	31·95

Another means of determining the weight of waste products is to weigh the fuel carefully and analyse the escaping gases. In outline the process is very simple. Analyse the coal and find its percentage of carbon $= p_1$; analyse the waste products and find the volumetric percentage of carbonic acid which they contain p_2. This is exactly equal to the volume of oxygen required to produce it. It has already been shown that if all the oxygen in a sample of air (viz. 20·8 %) has been consumed, the weight of the products of combustion is 12·587 times the weight of carbon burnt, so that

$$Q = q \cdot \frac{p_1}{p_2} \cdot 12{\cdot}587 \cdot \frac{20{\cdot}8}{100} = q \cdot \frac{p_1}{p_2}\, 2{\cdot}62.$$

Here q is the weight of fuel burnt per hour, Q is the weight of waste products per hour. Thus if $q = 2{,}000$ lbs. per hour, $p_1 = 80$ %, and $p_2 = 10$ %, the waste products per hour weigh $Q = 41{,}889$ lbs.

Influence of Hydrocarbons on the Waste Gases.—This result is only correct if the fuel contains no hydrocarbons, hydrogen, or oxygen. When these are present the sum of the percentages of oxygen and of carbonic acid contained in the waste products are not equal to 20·8 %, as they should be. It is therefore necessary also to determine p_3, the percentage of nitrogen, which in dry air is 79·2 %. Then

$$Q = q \cdot \frac{p_1}{p_2} \cdot \frac{79{\cdot}2}{p_3} \cdot 2{\cdot}62 = \frac{q \cdot p_1}{p_2 \cdot p_3} \cdot 207.$$

This formula is sufficiently accurate for all determinations required in practice. Where absolutely correct results are aimed at a thorough analysis of the fuel, particularly with regard to carbon, sulphur, hydrogen, oxygen, and nitrogen, has to be made. Sulphur consumes its own weight of oxygen, and the volume of the acid produced is exactly equal to the oxygen used. Therefore ·375 of the percentage of sulphur should be added to p_1, while p_2 would represent the sum of the volumetric percentages of the carbonic and sulphurous acids. Hydrogen uses up eight times its weight of oxygen and forms steam, which does not get measured in the analysis, while the oxygen and nitrogen contained in the fuel will be found there. To carry this out in detail would lead to complicated results which are of no use, as it will be found that the composition of the waste products varies from minute to minute. A rough estimate of the percentage of hydrogen, h, burnt at various periods can be made with the help of the following formula :

$$h = p_1 \frac{(p_3 - 79{\cdot}2)}{3 \cdot p_2}.$$

Being able to determine the percentage of carbon and hydrogen, the idea naturally suggests itself to use this analysis of the waste gases for the purpose of measuring the amount of heat produced in the furnace. Applying the numerical values of the heats of combustion (p. 62), the initial temperature of the burning fuel is found to be

$$t_1 = t_0 + p_3\,[p_2 2{\cdot}96 + 3{\cdot}59\,(p_3 - 79{\cdot}2)].$$

Here t_0 is the temperature of the air. The amount of heat in calories generated in the furnaces connected to one funnel is

$$H' = Q\,.\,(t_1 - t_0)\,{\cdot}237 = Q\,.\,p_3\,[{\cdot}7\,p_2 + {\cdot}85\,(p_3 - 79{\cdot}2)].$$

In this case Q has to be measured with the help of an anemometer (see fig. 106), and by substituting the values for Q and dividing by 966 the amount of heat generated can be expressed in evaporative units :

$$E = \frac{V\,.\,p_3}{T}\,39{\cdot}7\,[{\cdot}7\,p_2 + {\cdot}85\,(p_3 - 79{\cdot}2)].$$

So that if sufficient reliance could be placed on the readings of the anemometer and on the chemical analysis of the gases, it would be possible to ascertain the amount of heat generated in the furnaces of a boiler without having recourse to the tedious and uncertain processes of weighing and analysing the coal ; but until the two methods have been tried together and found to agree, perfect reliance cannot be placed on either.

Variations in the Process of Combustion.—It will be found that during the early stages of combustion sometimes twice as much hydrogen is burnt as during the latter stages, while the average agrees fairly well with the coal analysis. Thus in the case of the 'Iona' trial ('M. E.,' 1891, p. 208) the coal analysis gives 5·47, while the average of 13 gas tests gives 4·86 %, the minimum being 3·35 and the maximum 7·49. The other gases in the fuel were not determined.

Another interesting determination of waste products has been mentioned by Mr. Bramwell ('C. E.,' 1878, vol. lii. p. 156).

Volumetric Analysis of Gas in Flue. Coke Fires.

Time of Taking Samples	Before Firing	After Firing	Between Firing	Averages
Carbonic acid, per cent . .	9·42	12·91	10·19	10·84
Oxygen " . .	11·16	7·16	10·16	9·49
Nitrogen " . .	79·46	79·93	79·65	79·67
Ratio, waste products: carbon	22·5	16·4	20·8	19·55

The fuel was said to have contained 18·4 % water and ashes, so that $p_1 = 81{\cdot}6$ %. It will be noticed—and this is only natural to expect—that less excess air is passing through the fuel immediately after firing than at any other period.

An additional feature in this experiment is, that the quantity of air admitted to the fire was measured by means of an anemometer, and together with 194 lbs. of carbon burnt, made a total of 3,524 lbs. of waste products. The above chemical analysis would make it 4,660 lbs., or 30 % more. This is doubtless the more correct determination ; at

any rate it reduces the 'heat units unaccounted for' by 80 %, and the balance-sheet which is there published is now nearly correct.

Other gas analyses will be found in the following publications :— 'C. E.,' 1889, vol. xcix. pp. 60, 63, 64, 66. 'M. E.,' 1889, p. 235 ; 1890, p. 206 ; 1891, p. 208 ; 1892, p. 8. 'Enging.,' 1890, vol. l. pp. 59, 121, 122, 383, 413, 592, 593 ; 1891, vol. li. pp. 236, 237, 577 ; vol. lii. p. 375 ; vol. liii. p. 344.

The importance of obtaining accurate results as regards waste gases when making economic trials has now been sufficiently demonstrated ; but as no book with instructions how to make these experiments exists, it has been thought necessary to enter more fully into details than the subject would otherwise warrant. Mr. C. Wilson, who carried out these tests on the trials of the Research Committee, has kindly permitted sketches of various flasks, &c., which were used on these occasions to be reproduced here.

Collection of Gas Samples. In order to obtain really reliable results, great care has to be taken to obtain average samples of gas. It would be good if continuous supplies could slowly be drawn from a central tube of every smoke box, and each one analysed; or, if that is impossible, the supplies might all be brought together in one tube, and an average sample taken. At present they are drawn from the funnel, but a little consideration will show that if it is a large one (and some are 20 ft. in diameter) connected to many furnaces, and if the sample is taken through a straight pipe with numerous holes, it is likely to represent, not the average quality, but that of one particular furnace. This view is supported by the results obtained on the trials of the s.s. 'Colchester' ('M. E.,' 1890, p. 218), where the mean temperature in the forward funnel was 415° F., while the air supply was 16·1 lbs. per lb. of coal. In the aft funnel the values were 445° F. and 23·5 lbs. of air, which relations are the reverse of what might be expected. Pipes[1] should, therefore, be led to various points of the section—for instance, to A, B, C, D, E, F (fig. 107); or, if this is insufficient, more arms can be added, and also the centre of the cross may have an opening, and two or more should be fixed on each arm. Their position should be so chosen that the distribution is uniform.

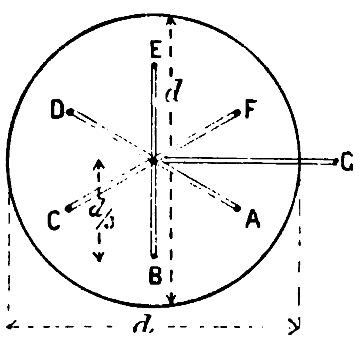


FIG. 107.

In order that equal amounts may be drawn from each point, the sizes of the holes should all be equal (fire-proof gas burners can be used), and the section of the tubes which connect them ought to be fairly large, so as to reduce the internal friction. Before bottling a sample a considerable quantity of gas has to be sucked through the tubes, so as to remove all trace of atmospheric air. If the sampling is to be a continuous one, the tube G (fig. 107) is now connected to a bottle filled

[1] For high temperatures iron pipes should not be used, because they effect a reduction of the oxygen in the waste gases.

with water. By means of a siphon, SS (fig. 108), the fluid W is drawn off into the lower bottle, and the upper part of the bottle G filled with the gas. At stated intervals the tube S is closed, stopping the flow of the gas. A flask, B (fig. 109), filled with water, is now attached to the tube *a* of the bottle W, and on raising D the collected gas, G, is

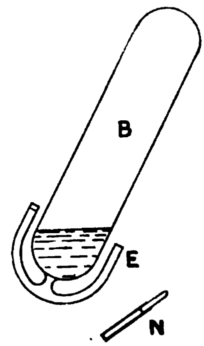


FIG. 109.

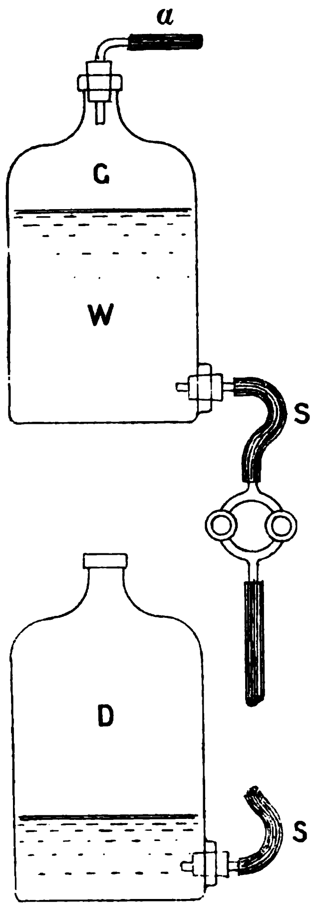


FIG. 108.

driven into B, the water running out of the opening E, which, as well as the other, is then closed by an india-rubber nipple, N. In the trials of the Research Committee mercury was used. The remainder of the gas G is then driven out by running more water from D into W, after which the process of collecting is repeated. The bottle B now contains the gas to be analysed, with a small quantity of water in its bottom end, which hermetically seals it and prevents any adulteration by the atmosphere.

A little forethought is necessary in arranging the periods of drawing the sample B from the bottle W. As it is being filled only very slowly—say, once per hour—and as the capacity of the tubes leading from the funnel is considerable, it is quite possible that after the first sampling it may take half an hour before the gas has travelled from one of the burners to the bottle W, so that the correct times of sampling have to be found by adding this interval to the periods of trials. Thus if the volume of the tubes, taking into account differences of temperature, is equal to the volume of fluid drawn out of W in, say, 30 minutes, it will be necessary to commence drawing the air slowly as soon as the trial commences ; but the gas which has been collected during the first half-hour should be blown

away, and the actual sampling should not commence till then, and should be continued for half an hour after the completion of the trial. The pipe G leading to the tube *a* of the bottle W should be arranged to slant downwards, so that the gas which it contains, and which is cooler and heavier than the rest, does not fall back again and get mixed with that which has just entered the burners.

If the samples have to be taken at definite periods, the bottles W and D are not necessary. B is connected direct to the tube G (fig. 107), but before filling it all the stagnant gases in this and the adjoining tubes have to be drawn off by a suction pump or bellows.

Analysis of the Gases.—Having filled the bottle B, it is connected to a graduated tube, *d d* (fig. 110), containing water, which is slowly

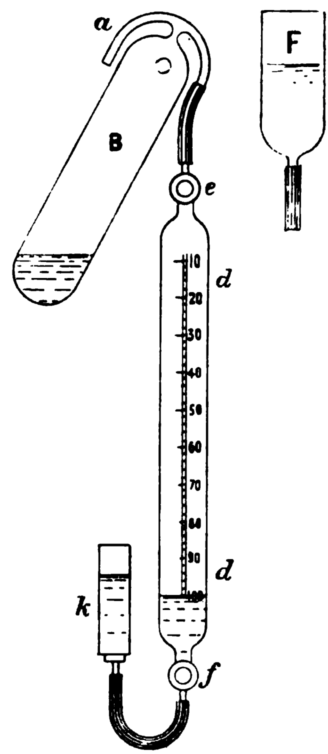


FIG. 110.

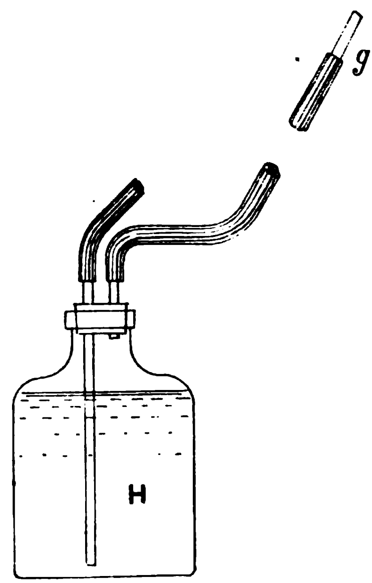


FIG. 111.

drawn off till the gas occupies as nearly as possible 100 divisions. Care must be taken during the analysis which has now to be carried out that no change of temperature or pressure occurs. The safest plan is to have the graduated tube permanently surrounded by a water jacket, in which a thermometer should be suspended. The correction for unavoidable changes is to subtract 1 % from the measured volume for every 5° F. or 3° C. of rise of temperature.

One of the nipples of B is removed under water, and the tube *a*,

communicating with a water bottle, is attached. The other nipple is then removed, and while the water is gently flowing through *a*, the connection to the cock *e* of the graduated glass tube *d* is effected. B is then turned down as shown in fig. 110, and the cocks *e* and *f* are both opened, until the fluid in *d* is lowered a little below the mark 100, as shown. Both cocks, *e* and *f*, are then closed and the bottle B removed. After allowing a little time for cooling—adjustment of temperature—a slight pressure is applied through *f* till the water level 100 is reached; then *e* is opened again, in order to dissipate the excess pressure.

The Fluids required for the Analysis are—

1st. A solution of caustic potash or soda, for the absorption of carbonic acid and sulphurous acid gas.

2nd. A concentrated caustic solution of pyrogallic acid, for the determination of the unconsumed oxygen.

3rd. A strong solution of copper subchloride in hydrochloric acid, for the determination of carbonic oxide.

4th. A solution of caustic potash, for the removal of the hydrochloric fumes of No. 3.

Wrong results are obtained if these chemicals are used in the wrong order. Thus No. 2 will absorb both oxygen and carbonic acid, and No. 3 absorbs oxygen and carbonic oxide.

Both the pyrogallic acid and the subchloride of copper must be carefully protected from the air, as they readily absorb oxygen and lose their powers.

The usual plan of operation is to attach the cock *e* to a closed bottle containing the chemical, and to draw the gas into it. After the absorption has taken place, the gas is drawn back again into the tube *d*, and the loss of volume is then measured.

A quicker plan is to pour the various chemicals into the tube *d*. This is done by attaching a funnel, F (fig. 110), to the cock *e*. The fluid is poured into it, and on opening both *e* and *f* it runs into *d*, the water in the bottom escaping. When F is nearly empty *e* and *f* are closed and the whole apparatus is shaken. Then a considerable quantity of water is poured through the tube *d*, and after standing five minutes to drain the reading is taken, and then the next chemical added and the process repeated.

Another plan, by which the chemicals are not wasted, is to extract the water or other fluid contained in the bottom of the *d* tube by suction. A bottle, H (fig. 111), containing water is attached to *f*. The air is drawn out of H by sucking at *g*, and as *f* is open the fluid in *d* descends; *f* is then closed. The bottle H is replaced by a similar one containing the correct chemical, which is drawn into *d* when *f* is opened. After being shaken it is drawn back into the bottle H; water is again admitted, readings taken, and the process repeated.

In order to ensure that the pressure in the *d* tube is the same as on filling, no reading should be taken without first attaching a glass tube, *k*, of about the same diameter as *d*, as shown in fig. 110. It is partly filled with water, and has to be moved up or down till the two water levels coincide.

The water used in all these operations should not be fresh, and particularly not recently boiled; otherwise it will absorb measurable

quantities of all the gases. A strong solution of salt in water slightly acidulated can be used with advantage, as it absorbs only small percentages of these gases. A good plan is to shake it up well with waste products of combustion before using it.

Mercury is often used instead, but though it does not absorb gases, it is more difficult to obtain correct readings, an error of one-hundredth of an inch with this fluid being equal to an error of about $\frac{1}{8}$ in. of water. Having by these means determined the composition of the waste gases, it is an easy matter to judgé whether the combustion was a perfect one.

It has already been explained how the weight of the waste products can be determined with the help of the known coal consumption and the known composition of the products of combustion, but it must not be forgotten that such an estimate would only be the average for the whole trial; where it is desired to make determinations at definite periods -viz. before, after, and between coaling—direct measurement of the volumes of gas must be made, which can be combined so as to give a mean value capable of being corrected by comparison with that determined by analysis.

Funnel Temperatures.—The chief object of either measurement is the estimating of the heat wasted, and this can now be easily done if frequent readings are taken of the funnel temperatures. In this case, too, care should be taken to get good average results, for it is quite conceivable that if the thermometer is fixed in one position it will only record the temperature of the gases from one particular furnace, which may be hotter or colder than the rest.

For a list of pyrometers see p. 73, to which should be added that for funnel temperatures; the most convenient one is an ordinary mercurial thermometer, made of glass which can stand high temperatures. Quartz would be better. The vacuum end of the tube is filled with nitrogen, which prevents the mercury from boiling. Such instruments were used on the trials of the Research Committee ('M. E.,' 1889-92), and registered up to 860° F. This was not quite high enough for some of the trials, but the sodium and potassium thermometer of E. C. C. Baly and J. C. Chorley is said to register up to 1,100° F.

Great care must be take to screen the thermometer bulb from radiant heat, and yet allow the air to come in contact with it; otherwise the readings will be wrong. Thus the temperature of waste gases will apparently be very much less at the top of the funnel than lower down, which is partly due to radiation into space of the heat acquired by the thermometer bulb, and partly due to the cooling of the gases as they travel upwards. In the same way the heated air which is admitted into some furnaces is incorrectly measured if this is done by an unprotected thermometer, because of the radiation, either from the hot fuel or from the heated iron. The screens should be at least twofold, and ought to be highly polished on their outer surfaces, so as to reflect all radiant heat.

CHAPTER IV.

HEAT TRANSMISSION.

Nature of Heat.—At one time heat and light were looked upon as being two different substances or conditions; now it is universally admitted that they are identical, the only difference being an objective one, and depending solely on the incapability of our optic nerves to perceive heat rays. Our other nerves make no distinction between the two, and the back of the hand will as readily feel the radiation from a blackened stove as from an electric arc, the one emitting only heat and the other chiefly light.

By those who believe in the existence of a universal ether, light and heat are thought to be vibrations of this medium. Every known experiment on the subject points to the conclusion that these vibrations are of a transverse nature, like waves on the surface of fluids, and no relation seems to exist between these and the longitudinal vibrations called sound. The lengths of the waves of visible heat (light) range from 30,000 to 60,000 per inch. The non-luminous heat waves are longer, and the ultra-violet light, invisible to our eye but very active in photographic reactions and in vegetable life, consists of very much smaller waves. The speed with which heat and light travel through space is at the rate of about 187,000 miles per second, so that the number of vibrations per second has to be counted by hundreds of billions.

Radiant Heat of Gases and Solids.—Heat being invisible, many of the phenomena connected with radiation will be more readily understood if exemplified by the action of light, though Prof. Langley, ('C. E.,' vol. c. p. 469) has done a good deal to confirm, by direct experiment, the conclusions which naturally suggest themselves when comparing light and heat.

It has been found that solids, when heated to incandescence, radiate heat and light of every colour, which is very clearly shown in the spectroscope by the fact that the spectrum of a luminous solid is a continuous one, light being visible continuously from the red end to the violet, and perceptible beyond both ends. Luminous gases do not radiate light of every colour, and some are even strictly monochromatic. Thus the vapour of the metal cæsium has a spectrum consisting of only one green line. Indium shows three lines, chiefly blue; hydrogen shows four lines, placed in various parts of the entire spectrum. Arsenic and sodium show several lines near the yellow; potassium also shows comparatively few lines. For further information see W. M.

Watts, 1872. Naturally all these gases also radiate definite colours of heat, if one may use this term; but the important point is that the light or heat thus radiated is infinitesimally small as compared with the heat radiated by a solid of the same temperature; so that it will take infinitely longer to cool a gas than a solid. An idea of the enormous disparity existing between the two will be obtained by examining the spectrum lines of sodium with a good instrument. They will appear to be no wider than very fine inked lines, whereas the full spectrum from red to violet is measured by yards, which represents the amount of radiation from a solid of the same temperature as the gas. The breadth of the lines represents the entire heat radiation of sodium vapour.

The radiating power of solids also varies considerably, as will be seen from the following table. The values for a carbon film were obtained from experiments kindly made for the author by Messrs. Slingo and Brooker on an incandescent lamp, and with an optical pyrometer by MM. Mesuré and Noël, which unfortunately is only graduated up to about 2,000° F. The experimenters found no appreciable differences in the results on immersing the glow lamp either in cold or in boiling water. The values for other substances are deduced from W. H. Preece's experiments ('Proceedings,' 1884 and 1887) on the heating effects of electric currents (see also p. 92):—

Evaporative Units of Heat Radiated per Square Foot of Exposed Surface per Hour per Degree (F.) of Difference of Temperature.

Temperature of object, °F.	169	980	1,600	1,900	2,200	2,500	3,000
Carbon filament in vacuo	—	—	·7	·6	3·7	3·5	3·2
Iron wire in air	7·4	4·5	—	—	—	—	—
Platinum wire in air	9·3	8·0	—	—	—	—	—
Copper " "	27·2	14·9	—	—	—	—	—
Aluminium " "	12·1	12·8	—	—	—	—	—

Absorption of Heat.—Gases absorb only such light as they can emit, but experiments as to the influence of temperature on this phenomenon are still wanting. According to present views it would appear that gaseous flames are only luminous at their outer surfaces, which is contrary to what one sees.

It may, therefore, be assumed that flames, unless they are quite luminous, radiate and absorb little heat, and none at all when they have grown cold and transparent. The heat which radiates from a glowing coke fire is not absorbed by the non-luminous flame over it. It also follows that when the gases are cooled below the point of luminosity they cannot transmit any of their heat to the boiler plates, except by actual contact, or—and this is of great importance—by contact with the suspended particles of soot and ashes, which, as they are solids, are capable of radiating heat even at low temperatures. It would therefore appear probable that a little smoke is a good admixture to the products of combustion. It is also probable that luminous flames, such as are produced from North-country coal, will be more efficient radiators in the furnace than those which are produced from coke, anthracite, &c. In the former case the radiating surface, i.e. the outer part

of the flame, is very much larger than the exposed incandescent surface of the fuel on the grate.

Another matter which should be mentioned here is the influence of moisture and of carbonic acid on the radiating power of air. Prof. Tyndall (1870, pp. 320, 359) states that carbonic acid absorbs (and therefore also emits) 972 times as much heat as any of the permanent gases, including oxygen and nitrogen, and that humid air (he does not give the percentage of moisture) absorbs 90 times as much heat as dry air. As the products of combustion contain from 10 to 20 % of carbonic acid, this alone would raise the absorbing and radiating power about one or two hundredfold, and the moisture contained in the air and in the fuel would add considerably to this power. As these impurities, including smoke, vary considerably in every trial, it cannot be expected that much uniformity will be obtained in experiments which do not take them into account.

Of more importance than the theories as to how the heat transmission takes place, is the question of how much heat can be transmitted. This very soon resolves itself into the oft-debated subject as to the relative merits of the heating surfaces of the furnaces and of the tubes. Unfortunately few experiments have been carried out, and those few are very incomplete.

De Pambour was the first to raise the question ('Comp. Rend.,' 1840, vol. x. pp. 32, 111, 480) and to make comparisons, but these are valueless.

C. W. Williams ('Engineer,' 1858, vol. v. pp. 223, 243, and 'N. A.,' 1862, vol. iii. pp. 122) made experiments on the heat transmission of tubes.

J. Graham ('Manch. L. Ph.,' 1860, vol. xv. p. 8) experimented on the heat transmission of flat surfaces.

M. Geoffroy ('Couche,' 1877, vol. iii. p. 28) made experiments on a locomotive boiler whose length was subdivided.

P. Havrez ('An. Génie,' 1874, 2nd ser. vol. iii. p. 520) and J. A. Longridge ('C. E.,' 1878, vol. lii. p. 101) give analyses of these experiments.

Reference is also repeatedly made to M. Petit's experiments, but details are wanting. They seem to have consisted of two series : in the one lot coke was burnt, and in the other briquettes ; and the draught was $\frac{3}{4}$ in. in the one case and 4 ins. in the other. Compare P. Havrez (see above), p. 550, and 'Chron. Ind.,' 1873, vol. ii. p. 425.

C. W. Williams ('Engineer,' 1858, vol. v. p. 206) also mentions experiments on a subdivided boiler carried out by Messrs. Wood and Dewrance, but all details are wanting.[1]

Tube-Heating Surface.—Of the above-mentioned experiments those by Mr. Williams are the most complete, as far as he attempted to go ; but the conditions are such that the external radiation must have been out of all proportion to the heat received by the solitary internal pipe. Some of the compartments could not even be raised to the boiling point of water. All the other experiments are still more unsatisfactory, as will be found when examining them.

M. Geoffroy's experiments, of which the results are reproduced in D. K. Clark's work (1890), have been analysed by the author, but

[1] Since writing the above very valuable experiments have been published by A. J. Durston, *N. A.*, 1893, vol. xxxiv. p. 130.

under great difficulties, for no mention is made of the steam pressure, except that it is high ; no statement can be found whether the boiler-shell plates were lagged ; the calorific value of the fuel is not given, nor the temperature or weight of the waste products. A very cursory examination showed that experiments Nos. 5 and 10 differed too much from the others to be relied upon. By trials and corrections of errors it was found that the ratio of products of combustion to fuel burnt must roughly have been as follows :--

Experiment No.	1	2	3	4	6	7	8	9
Ratio	27	27	19	19	22	21	22	22

It was also found that E, the weight in lbs. of water evaporated per square foot of heating surface per hour, can be estimated by the formula

$$E = ct^n, \text{ or } \log E = \log c + n \log t.$$

Here t is the difference of temperature between the boiler water and the hot gases at any particular point of the tubes, and c and n are constants. The closest agreement with the whole number of experiments is obtained by assuming that the calorific value of the fuel is $12\frac{1}{3}$ lbs. of water evaporated under the conditions of the trial, or, say, 15·1 lbs. from and at 212° F., further that $\log c = -5{\cdot}4$, $c = \frac{1}{250000}$, and that $n = 2$. Then

$$E = \frac{t^2}{250000},$$

and e, the units of heat transmitted, is found by multiplying E by 1,150, viz.

$$e = \frac{t^2}{217}.$$

In order to obtain an idea of the distribution of the evaporation along the lengths of the tubes, it will be necessary to remember that while the gases move on, they are being gradually cooled. The amount of cooling, or drop of temperature, depends on E or e, the amount of heat abstracted, and on the velocity with which Q, the weight of air, passes over 1 square foot of tube surface per hour.[1] We have

$$\Delta t = -\frac{e}{Q\sigma} = -\frac{E.1150}{Q.\sigma} = -\frac{t^2.1150}{250000.Q.\sigma}.$$

Here Δt is the drop of temperature of the gas which passes over 1 square foot of heating surface, σ is the specific heat of the gas—say, ·237 ; so that

$$\Delta t = -\frac{t^2}{Q}\cdot\frac{1}{51{\cdot}5}.$$

The integral of this equation is

$$t = t_0 \cdot \frac{51{\cdot}5 . Q}{51{\cdot}5\, Q + S . t_0}.$$

[1] The heat transmission also seems to be affected by the velocity of the gases. See pp. 78–82, 98.

Here t_0 is the initial and t the final difference of temperature, and S the heating surface of the tubes in square feet.

The quantity of water evaporated by this heat, which is abstracted from the gases, is

$$E = \frac{Q \cdot \sigma\,(t_0 - t)}{1150} = \frac{t_0 \cdot Q \cdot \cdot 237}{1150}\left(1 - \frac{51 \cdot 5\,Q}{51 \cdot 5\,Q + S \cdot t_0}\right).$$

In M. Geoffroy's experiment No. 1 it has been assumed that the products of combustion are 27 times as heavy as the coke burnt. $Q = 27 \times 436 = 11{,}772$ lbs. of waste gas per hour. The heating surface of each compartment of tube surface being 179 square feet, and the theoretical evaporative value of the fuel having been assumed to be $12\frac{1}{3}$ lbs., the temperature of the flame as it leaves the fuel would be

$$T = \frac{12\frac{1}{3} \cdot 1150}{27 \cdot \cdot 237} = 2{,}220^\circ \text{ F. (see p. 71).}$$

To this has to be added the initial temperature of the air, and the temperature of the boiler water has to be subtracted, leaving, say, 1,980° F.

A further subtraction has to be made on account of heat lost in the fire box; this, according to the experiment, is 710° F.; so that the initial temperature of the gases which enter the tube plate is $t_0 = 1{,}270^\circ$ F. in excess of the water temperature.

The quantity of water evaporated in one, two, three, and four compartments of tube surfaces of the experimental boiler can now be calculated.

$$E = 1270 \cdot \frac{11772}{1150} \cdot 237\left(1 - \frac{51 \cdot 5 \cdot 11772}{51 \cdot 5 \cdot 11772 + n \cdot 179 \cdot 1270}\right)$$

$$= 3070\left(1 - \frac{1}{1 + n \cdot \cdot 375}\right).$$

Here n is the number of compartments, and we have

$$\begin{aligned} E_1 &= 3070\,(1 - \cdot 727) = 838 \text{ lbs.} \\ E_2 &= 3070\,(1 - \cdot 571) = 1{,}316 \text{ lbs.} \\ E_3 &= 3070\,(1 - \cdot 470) = 1{,}628 \text{ lbs.} \\ E_4 &= 3070\,(1 - \cdot 400) = 1{,}842 \text{ lbs.} \end{aligned}$$

The evaporations in pounds per hour in each compartment are the differences of these values, viz.

838 lbs., 488 lbs., 312 lbs., 214 lbs.;

but the experiments show

996 lbs., 430 lbs., 228 lbs., 128 lbs.,

and the differences are:

−158 + 58 + 94 + 76.

Priming, which is mentioned, would account for the large experimental readings in the first compartment, while the other differences are accounted for by radiation. The external surface of each cylinder being 31 sq. ft., the mean amount of radiation is equal to $2\frac{1}{2}$ lbs. of steam condensed per sq. foot per hour. This is not an excessive amount (see p. 97). The calculations which have been carried out on the

other experiments lead to similar results, so that until more accurate experiments are made this formula may at any rate be used for comparisons.

The above investigation shows that experiments are urgently needed. It also shows that the statement, that the value of tube surface for heating purposes is a definite fraction of that of the combustion chamber plating, is quite wrong. It is simply a question of temperature, and if the heat is not taken out of the flame in the fire box it will be taken out of it by the tube surface. If the gases are hot enough the tubes may be as effective as the thicker plates, if not more so, as will be seen from the following values calculated from the previous formula :—

Excess temperatures of gases over water . . . °F.	2,000	1,500	1,000	500	300	200	100
Pounds of water evaporated per square foot per hour . .	16·0	6·0	4·0	1·0	0·36	0·16	0·04

These few figures give an indication of the relative heating value of the last few feet of tube-heating surface as compared with the radiating surface of the shell. If, for instance, the gases on nearing the ends of the tubes have been cooled down to 200° above the water temperature, then every extra square foot of heating surface will only supply $\frac{1}{6}$ lb. of steam per hour, and this small performance may be too dearly paid for, particularly if the boiler is not lagged.

Furnace-Heating Surfaces.—A further very important question is the value of that heating surface which is exposed to the direct action of the flame and to the radiation from the fire.

Geoffroy's experiments would be exceedingly valuable for the purpose of this determination if he had only measured the temperatures, &c. J. A. Longridge ('C. E.,' 1878, vol. lii. p. 101) assumes it to be proportional to the difference of temperature, and fixes it at 11 units for every degree, which works out to about 22 lbs. of water evaporated per square foot per hour with good fires, but in some of M. Geoffroy's experiments it exceeds 40 lbs.

On the other hand, most text-books adopt Dulong and Petit's views ('An. Ch. Ph.,' 1817, vol. vii. pp. 113, 225, 337), that the cold as well as the hot object radiates heat at a rate which is expressed by an exponential function c^t, where c is a coefficient and t the temperature. The amount of heat absorbed by furnace plates would then be $c^a - c^b$, where b and a stand for the respective temperatures of the boiler plate and of the fire. These formulæ are constructed on the basis of experiments in which b ranged from 32° to 140° F., and a from 140° to 500° F., and, as more recent experiments do not confirm these, there is no need to employ a formula which is very complicated in its working. A. E. Kennely ('Elect. W.,' 1889, vol. xiv. p. 374) and N. Barbieri ('Elect. Z.,' 1891, p. 27) both experimented on this subject by heating wires by means of electricity, and measuring their temperature by means of their elongation. The temperatures do not exceed 350° F. Roberts Austen ('M. E.,' 1891, pp. 565, 590, plate 119) gives the photographic record of the cooling of a red-hot steel ingot. The amount of

heat lost per hour per square foot of surface amounts to about 50 evaporative units for a difference of 1,500° F., or at the rate of about ·03 evaporative unit per degree of difference of temperature. Below 1,000° F. the radiation seems to be reduced to about ·015 per degree of difference. These estimates are based on the assumption that the specific heat of iron is the same at both these temperatures, though there is evidently a marked change between the two. No great reliance can, therefore, be placed on these deductions. But that there is also a change in the rate of cooling, depending on the temperature, will be evident by analysing the case of the cooling of a railway axle between the temperatures of 32° and −40° F., mentioned in the discussion on the same paper ('M. E.,' 1891, p. 590, plate 123). There the rate of heat transmission has dropped to about ·0018 evaporative unit. Information on this subject, and particularly on the radiating power of luminous and non-luminous flames, could be obtained by measuring their temperatures, their sizes, and the heat generated or fuel consumed. Hot flames will be found smaller than cold ones, and for the same temperatures their exposed surfaces should bear some relation to the rate of combustion. That the size of the flame affects the evaporation from the furnace plates is shown by Geoffroy's experiments. It did not increase in proportion to the fuel consumed, showing that there are limiting conditions ; and it would appear that these are, that the heating power of a coke fire depends, not so much on the exposed heating surface, as upon the grate surface, and that after a certain point has been reached, the heating power of a grate covered with coke is independent of the amount of fuel consumed. Secondly, block fuel, which gives off a luminous flame, radiates nearly 50 % more heat in the fire box than coke does, indicating that the size of the flame, or perhaps its nearness to the plates, is an important factor. It will also be found, that with this fuel increased consumption causes increased evaporation in the fire box, showing that the flame had either increased in size or, on account of its higher temperature and density, had grown more effective.

In these trials the consumption varied from 50 lbs. to over 100 lbs. of fuel per square foot of grate per hour, and the evaporation ranged from 20 lbs. to 40 lbs. of water per foot of fire-box heating surface per hour. It will be of interest to compare these values with those obtained by other experimenters.

M. Hirsh ('Soc. d'Enc.,' 1890, vol. v. p. 30) carried out two sets of experiments for determining the evaporation ; in the first he bolted a small cylinder to the water side of the heating surface, directly over the fire bridge. It was sufficiently high to reach well above the water level, and was connected to a gauge glass by means of a small pipe which was also used for feeding purposes. When the boiler was at work this cylinder was constantly replenished with water, but while readings of the gauge glass were taken the feed was cut off. The coal consumption per square foot of grate varied from 17 lbs. to 53 lbs. per hour, and the evaporation varied from 26 lbs. to 62 lbs. per square foot per hour from and at 212° F. The best results were obtained when the grate was burning about 40 lbs. per hour, which shows that the experiments are not as reliable as could be wished ; but if better arrangements were to be made for ensuring circulation, similar

to those employed in Field tubes, it is possible that the absolute value of the various parts of boiler heating surfaces could be directly obtained.

In Graham's experiments the evaporation did not exceed 20 lbs. per square foot per hour, and Stephenson found it to be $16\frac{2}{3}$ lbs.

Indirectly, the experiments detailed by A. F. Yarrow ('N. A.,' 1891, vol. xxxii. p. 108) can be used for determining the amount of heat transmitted through a plate. He measured the curvature of a tube plate which was covered with water and heated over a smith's fire. Its acquired radius was 550 inches, i.e. one of its surfaces had expanded $\frac{1}{733}$ of its length more than the other side (assuming the plate to have been 1 in. thick). But this can only have been brought about by the fire side of the plate being 330° F. hotter than the other, and as experiments made on the flow of heat in iron bars show that this difference of heat per inch of distance can only exist if sufficient heat is being transmitted to evaporate 142 lbs. from and at 212° F., this must have been the evaporation under these conditions.

A. F. Yarrow also made similar comparative experiments on iron and copper plates placed over gas jets, and found that 57 lbs. per square foot were evaporated over the iron plates, and only 32 lbs. over the copper plates, which is in accordance with other experiments. (See p. 98.)

Mean Temperatures of Heating Surfaces.—Another set of interesting experiments were carried out by J. Hirsh ('Soc. d'Enc.,' 1890, vol. v. p. 302), and are mentioned in the second part of his paper. He constructed a small kettle about 10 ins. in diameter, which had an iron bottom about $\frac{3}{8}$ in. thick. Arrangements were made for keeping the

Conditions	Water Evaporated from and at 212° F. per Sq. Foot per Hour	Melting Temperature of Alloys		
		Above	Near	Below
	Lbs.	°F.	°F.	°F.
Distilled water	26·8	...	335	...
	35·6	...	369	...
	64·5	...	428	...
	82·5	428	...	482
	93·5	"	...	"
Water which contained ½ % starch	35·8	369	...	428
	59·2	...	428	...
	77·0	428	...	482
Distilled water. $\frac{1}{25}$ in. scale on plate	33·8	369	...	482
	46·1	...	428	...
Distilled water. $\frac{1}{5}$ in. scale on plate	34·6	428	..	482
	53·2	...	842	...
Distilled water. Plate greased with mineral oil	33·8	...	428	...
	34·3	...	428	...
	47·1	...	428	...
	62·6	482	...	635
	63·2	..	...	"

water level at a constant height, while a strong gas and air blast was directed against the bottom, of which a small area of 4 ins. in diameter was exposed. Twenty-four holes were drilled into the bottom of this plate, $\frac{1}{8}$ in. in diameter and $\frac{1}{4}$ in. deep, and these were filled with lead and tin alloys. In all 38 experiments were carried out, of which the results of a few are given in the table on p. 94.

In order to compare these values with the estimated transmission in A. F. Yarrow's experiments, the excess temperatures above 212° must be divided by $\frac{3}{8}$, that being the ratio of the thicknesses of plates, and it will then be seen that in this case the heat transmitted is less, but grave doubts are entertained whether the indications of the alloys can be relied upon. Their outer surface is not iron, and they are not soldered to the bottoms of the holes, as they should be, and according to some supplementary experiments, in which two thicknesses of plate were bolted together, the resistance to the passage of heat across the boundary of the two metals will have been very great, and the indicated temperatures must therefore be looked upon as excessive.

This is particularly the case where the plate has been covered with scale, and it is of interest to note that when this is $\frac{1}{5}$ in. thick the temperature of the fire side of the iron plate has to be increased an extra 460° when evaporating 55 lbs. of water per hour. According to this experiment scale offers about five times as much resistance to the passage of heat as iron does, whereas laboratory experiments show the ratio between iron and plaster of Paris to be as 1 to 10. This may be due to the thermal conductivity of iron having been measured along the fibre, while in these experiments the heat travelled across the plate and across the various layers of fine slag.

A very important point to be noted is, that even a $\frac{3}{8}$-in. plate can be heated to above the melting temperature of lead if coated with a little scale on the water side, provided the fire is so hot that 55 lbs. of water are evaporated per square foot per hour, and it is therefore not unreasonable to suppose that most of the recent troubles with the tube plates of Navy boilers are due to over-heating, particularly when it is remembered that it is probably not water, but moist steam which is in contact with these plates.

The last set of experiments also show that an injurious effect is obtained by allowing grease to settle on heating surfaces.

Somewhat similar experiments have recently been made by the late Dr. Kirk ('Enging.,' 1892, vol. liv. p. 333.) As in the above case, plugs of alloys were fitted into the bottom of a plate for determining its temperature. It was originally $2\frac{3}{4}$ ins. thick, and was gradually reduced to $\frac{13}{16}$ in. As no measurements were taken of the water evaporated, the results are of less value than the above. A. J. Durston ('N. A.,' 1893, vol. xxxiv. p. 130) has also made valuable experiments on this subject, but did not measure the evaporated water.

Thermal Conductivity.—Various determinations of the coefficients of transmission of heat will be found in the following table, which has been calculated from values collected in Everett's 'Units and Physical Constants':—

Thermal Conductivity of Solids.

Materials	G.C.S. Scale	English Scales: Thermal Units	English Scales: Evaporative Units	Observers
Iron at 32° F.	·207 to ·154	603 to 449	·621 to ·462	Forbes
„ at 212° F.	·157 to ·129	456 to 375	·472 to ·387	„
„ at 527° F.	·124 to ·112	361 to 297	·372 to ·306	„
„	·164	477	·492	Neumann
„	·199 (1 − ·0029 t° C.)	610 (1 − ·0015 t° F.)	·6 (1 − ·0015 t° F.)	Ångström
Copper	1·027 (1 − ·0021 t° C.)	8140 (1 − ·00115 t°F.)	3·08 (1 − ·0021 t° F.)	„
„	1·108	3220	3·024	Neumann
Brass	·302	878	·906	„
Zinc	·307	892	·921	„
German silver	·109	317	·327	„
Slate, along cleavage	·0055 to ·0065	160 to 190	·0165 to ·0195	„
„ across cleavage	·00315 to ·0036	92 to 105	.0095 to ·0108	„
„	·0081	23·5	·0243	Forbes
Clay, sun-dried	·00223	6·6	·00669	Neumann
Chalk	·0020 to ·0033	5·8 to 9·6	·006 to ·010	„
Fire-brick	·00174	5·1	·00522	„
Plaster of Paris, wet	·00164	4·8	·00492	„
Coal	·00057 to ·00113	1·65 to 3·3	·0017 to ·0034	„
Pumice stone	·00055	1·60	·00165	„
Various woods	·00026 to ·00359	·76 to 1·71	·00078 to ·00177	Peclet
Caoutchouc	·00041	1·19	·00123	„
„ vulcanised	·000089	·258	·000267	Forbes
Guttapercha	·00048	1·40	·00144	Peclet
Powdered charcoal	·00022	·64	·00066	„
„ coke	·00044	1·28	·00132	„
Charred wood	·000122	·35	·000366	„
Grey paper	·000094	·273	·000282	„
Pasteboard	·000453	1·33	·00136	Forbes
Paraffin	·00014	·41	·00042	„
Flannel	·0000335	·097	·0001	„
Water	·00136	4·00	·004	Weber

In this table the values given in the column headed 'Thermal Units' are the number of units of heat which one square foot of heating surface 1 in. thick will transmit per hour, if the difference of temperature of the two surfaces of the plate itself is one degree Fahrenheit. The values in the column 'Evaporative Units' are found from the last by dividing them by 966, which is the number of thermal units required to evaporate one pound of water from and at 212° F. These values can also be obtained direct from the G.C. S. values by multiplying them by 3.

It should be mentioned that all these experiments have been carried out on rods or rings, and that they are not absolutely reliable, because they are based on an imperfect knowledge of the changes of the specific heat and of the radiating power of the substances. If carried out on a plan similar to the one adopted by A. F. Yarrow for showing the curving of heated plates, these difficulties might be overcome, and, what is of almost greater importance, the coefficients of transmission of heat across the fibre of the material could thus be measured.

The relation between the thermal and electric conductivity is generally thought to be one of simple proportion, but H. F. Weber ('Acad. Berlin,' 1880, p. 457) has shown that

$$\frac{T}{E} = (\cdot 0877 + \cdot 136\,\sigma)\,10^4,$$

where T is the thermal conductivity, E the electric conductivity, and σ the specific heat of the substance.

Transmission of Heat.—Of more practical importance than the coefficients of conductivity are the rates of heat transmission, and the following table contains the results of some interesting experiments, in which one side of the plate or tube was exposed to hot steam and the other side to the atmosphere.

Coefficients of Heat Transmission.

Material and Conditions	Difference of Temperature	Steam Condensed per Square Foot per Hour	Thermal Units transmitted per Hour per Square Foot per Degree of Difference	Observer
	°F.	Lbs.		
Locomotive boiler	213	1·611	7·5	Isherwood [1]
$\frac{5}{16}$-in. iron plate	...	...	2·93	'Franklin Inst.' [2]
" " covered with cow-hair $\frac{1}{4}$ in.	...	...	1·05	" "
Do. do. $\frac{1}{2}$ in.	...	...	·57	" "
" " $\frac{3}{4}$ in.	...	...	·41	" "
" " 1 in.	...	...	·31	" "
" " $1\frac{1}{4}$ in.	...	...	·27	" "
" " $1\frac{1}{2}$ in.	...	...	·25	" "
Cast-iron pipe	...	·71	4·75	Meunier [3]
Wrought-iron pipe	...	·80	5·32	"
Copper pipe	...	·575	3·83	"
Cast-iron pipe	226	1·04	6·27	Meunier [4]
Wrought-iron pipe	222	1·04	6·42	"
Copper pipe	227	·828	4·96	"
Lagged cast-iron pipe	234	·465	2·67	"
" wrought-iron pipe	217	·405	2·58	"
Lagged copper pipe	229	·541	3·20	"
Ribbed cast-iron pipe	203	1·84	8·75	E. Deny [5]
" " "	210	...	1·04	H. Fischer [6]

Comparing these values with those obtained for the transmission of heat through tube surfaces, it will be noticed that the external surface of a boiler is about ten times more efficient as a heat dissipater than the very much thinner tubes are as heat absorbers (see p. 92). This is of importance, because it may happen that a boiler is made so long that the last foot of tube length supplies less heat than is given away by the

[1] Isherwood, *Experimental Researches*, vol. ii.
[2] *Franklin Inst.*, 1878, iii. vol. lxxv. p. 153.
[3] W. Meunier, *Rev. Ind.*, 1884.
[4] Ibid., *Soc. I. Mul.*, 1879, p. 730.
[5] E. Deny, ibid. 1883, vol. liii. p. 575, and 1884, vol. lv. p. 15.
[6] *Dingler's J.*, 1878, vol. ccxxviii. p. 1.

last foot of shell. Generally the entire circumference of the boiler shell is about one-fourth to one-sixth of the sum of the circumferences of the boiler tubes, and if this boiler is unlagged there would be no advantage in allowing the gases to escape at any temperature less than about 400° F. above that of the water in the boiler, or say 750° F. If properly lagged the gases may be cooled much lower.

In the above-mentioned experiments heat was supplied to the plates either by steam or hot water, and the cooling took place in air. No direct measurements have yet been carried out to ascertain whether the chief resistance is encountered on the hot or cold side. This could best be done by repeating A. F. Yarrow's experiments on the curvature of heated plates, and estimating the mean temperature of the plate by means of its lineal expansion. Some light might be thrown on the subject by comparing the previous experiments with the following (W. S. Hutton, 1887, p. 253). The units of heat which a $\frac{1}{8}$-in. plate will transmit per square foot per hour, if supplied with an unlimited amount of water on one side and steam on the other, are there given :—

Cast iron . .	265 units	Phosphor bronze	162 units
Wrought iron .	252 ,,	Copper . . .	155 ,,
Steel . . .	246 ,,	Tin plate . .	142 ,,
White metal .	207 ,,	Glass plate .	259 ,,
Brass plates .	175 ,,	Tiles . . .	246 ,,
Gun metal . .	168 ,,		

Another interesting feature of W. S. Hutton's figures is that they show that the transmission of heat is less dependent on the conductivity of the material than on its other properties. Thus tiles and glass plates are nearly as good as cast iron, while copper and tin plates are the least efficient.

Kirkaldy's corrugated evaporator tubes are said to transmit about 600 units. G. A. Hagenau's ('C. E.,' 1884, vol. lxxvii. p. 311) experiments are very exhaustive, and were undertaken to show the influence of the steam temperature, the water temperature, and the velocity of the cooling water. Unfortunately the results disagree amongst themselves, and it is to be feared that the precautions taken to guard against the presence of air were not sufficient, and possibly the external radiation may have varied. However the general results are, that the units of heat transmitted per square foot per hour per degree of difference of temperature sometimes amount to as much as 600.

Comparing even the lowest of all these values with those for the transmission of heat from metal to air, we find such a very great difference that there should be no hesitation in accepting the conclusion, that water is far more efficient than air in abstracting, and therefore also in imparting, heat.

Some meagre results are to be gleaned from the performances of steamers fitted with Howden's system of hot-air draught, according to which about $2\frac{3}{4}$ units of heat are transmitted per hour through one square foot for every degree of difference. The amount would be double for each surface on the assumption that the metal acquires the mean temperature. Plates which are exposed to the direct action of the flame also receive heat by radiation. (See p. 89.)

From the foregoing it is evident that our knowledge of the transmission of heat is very limited, and possibly incorrect ; nevertheless the

collection of previous experiments into one chapter has the advantage of showing in what direction further information should be sought, and the following experiments readily suggest themselves :—

1st. Experiments on the flexure of plates which are being heated on one side either by water, air, or radiant heat, and are being cooled on the other side by any of these methods.

2nd. A repetition of M. Geoffroy's experiments on a subdivided boiler, with accurate measurements of the temperatures of the waste products of combustion at various points, and also analyses of the gases, and calorific determinations of the fuels.

Temperature of Fire Bars.—The parts of the boiler which suffer most from the effects of slowness of transmission of heat are the fire bars. They are exposed to an intense heat at their upper surface, to radiation on part of their side surface, and the only available means of cooling them is by the air which passes over their sides. When the fires are thick, when much air is admitted above the bars, and when the draught is forced, the bars are naturally exposed to the very serious danger of overheating.

Assuming that the heat received by the upper surface of the bars is at the rate of 50,000 thermal units per hour per square foot, and assuming that this heat is being transmitted to the passing air at the high rate of ten units for every degree of difference of temperature. If a is the width of the air space, h the depth of the fire bar, b its breadth, and t its excess temperature over that of air, then

$$50000 \,.\, (b + a) = 10 \,.\, t \,.\, (b + 2h),$$
$$t = \frac{5000 \,.\, (b + a)}{(b + 2h)}.$$

In order to keep the temperature t below 1,000° F. it would be necessary to make $h = 2 \,.\, b + 2\frac{1}{2}\, a$, which is perhaps fairly near the truth ; but this leaves out of account the quantity of air, which is certainly an important factor. With ordinary draught the air supply is at the rate of about 400 lbs. per hour per square foot of grate, or about 7 lbs. per minute, which is certainly not a large quantity. Comparing the temperatures t_1 and t_2 in two different sets of fire bars, in which the various dimensions are a_1 and a_2, b_1 and b_2, h_1 and h_2, and the air supply Q_1 and Q_2, we have

$$\frac{t_1}{t_2} = \frac{b_1 + a_1 \,.\, (b_2 + 2h_2) \,.\, Q_2}{b_2 + a_2 \,.\, (b_1 + 2h_1) \,.\, Q_1}.$$

This would show that the more work a grate has got to do the deeper ought the bars to be made, and the greater should be their number. But as it is at first not so much the danger of burning the bars as of their being bent which has to be guarded against (see p. 10), making them very thin without reducing their length would increase this trouble. Until their temperatures have been taken under varying conditions, which is not difficult, it is idle to speculate any further on their behaviour, and at any rate the above formula should be looked upon more as an indication as to how much information is yet wanted, than as a practical guide for new departures.

CHAPTER V.

STRENGTH OF MATERIALS.

In this country, especially during the last ten years, steel has almost entirely supplanted wrought iron for ship and boiler construction. No further excuse is therefore needed for discussing its peculiarities more exhaustively than those of the older metal. Besides, the two materials are so intimately related, that the large amount of research expended on one must assist in explaining the other.

Wrought Iron.—It is usually stated that steel is a homogeneous material, while wrought iron is a conglomerate of crystals, granules, or fibres cemented together by films of slag. Layers and threads of slag undoubtedly exist in iron, but it is unreasonable to suppose that they are as uniformly distributed as the above theory would make it appear. The manner of the production of wrought iron in the puddling furnace seems to be, that drops of molten pig are slowly converted into plastic wrought iron, their centres always retaining a slight excess of carbon and other impurities, while the flame and slag acting on their outer surfaces are converting them into almost pure iron, possessing, amongst other qualities, that of welding with the greatest ease. That these numerous drops of white-hot metal, which may now be called granules, or even lumps, should stick together is but natural, and the trouble to be expected—and it does exist—is that numerous cavities filled with slag will come into existence.

When drawn out into bars or plates wrought iron would therefore consist of numerous fibres whose individual outer surfaces are very pure and soft, and whose cores contain the small percentage of carbon which had been allowed to remain. Professor Wedding has shown how by the microscope we can distinguish between the hard and soft part, for he found that the greater the percentage of carbon, the darker the colour if the iron or steel is raised to a blue heat; and by carefully polishing and etching samples of iron, and then heating them sufficiently to make them appear of a uniform dark straw colour, the microscope will show that this uniformity is an illusion, and that the metal really consists of innumerable cells of soft iron surrounding hard cores.

Influence of Producing Temperatures.—Had the temperature of the puddling furnace been as great as that in any of the steel furnaces, each granule would have been melted, and the product would have been mild steel of the same tenacity as the puddled iron; but it is well known that the first of these metals requires quite twice as much horse-power for machining it as iron of the same tenacity, and there does not seem

to be a better explanation than that this is due to the somewhat higher temperature at which it has been produced.

Steels from various makers, but of the same tenacity, are said to show great differences as regards the power required to chip them. An explanation for this might be sought for in variations in the casting temperatures, and these again would depend on the firebrick lining used.

J. W. Cabat ('American Inst. Mining Engs.,' vol. xiv. p. 85) deals with the influence of casting temperatures, and remarks that cold-blown charges of rail steel work better than hot-blown ones, and that open-hearth spring steel is similarly affected by the furnace temperature.

The Basic Bessemer Steel Process, using Phosphoric Pig Iron, and usually called the Thomas Gilchrist Process.—The pig is run into the converter in a molten state, and subjected from below to the action of a strong blast, which first removes the carbon and silicon, and then attacks the phosphorus. It was at one time believed that the development of heat in all the Bessemer processes was due to the burning of the carbon contained in the pig, but this has been disproved. Dr. F. C. G. Müller ('Deut. Ing.,' 1878, vol. vi. p. 387) mentions that silicon burns away first, and that for every per cent. consumed the temperature of the molten metal is raised 540° F.; that the carbon is not consumed until a temperature of 2,550° F. has been reached, and that its burning does not raise it. Phosphorus, like silicon, adds much heat to the bath, but will not burn until practically all the silicon and carbon have been consumed. It has also been found that the phosphorus cannot be consumed unless lime is present in the converter, and as this would attack and melt the ganister lining, which is nearly pure silicic acid, it is necessary to give the converters a *basic lining* (dolomite), from which the steel made by this process derives its name of '*basic steel.*' Only very little silicon may be tolerated in the pig intended for this process, as it attacks the dolomite, and in order to obtain a sufficiently high temperature, which cannot be done by burning carbon, phosphorus must be present in large proportions, viz. from 2 to 3 %. In some pigs phosphorus is not sufficiently plentiful, and then, in order to obtain the right heat, it has to be melted in Siemens-Martin furnaces instead of in cupolas.

During the process of manufacture the blast is kept up until all the phosphorus is consumed, and the metal in the converter is then almost absolutely pure iron. Spiegel and ferro-manganese are now added in the right proportions, and the charge is ready for casting.

At present the only available means for judging of the purity of the molten steel is to count the number of revolutions of the blowing engines from the time when the carbon lines in the spectrum have disappeared, but, in spite of assertions to the contrary, it does not appear that sufficient reliance can be placed on this proceeding, which is as follows:—

The composition of the pig iron is ascertained from samples if bought, or from daily returns if run direct from the blast furnace. The number of revolutions of the blowing engine required for supplying all the air necessary to consume all the carbon and silicon are estimated, and also how many extra revolutions will be required to remove the phosphorus. The weight of the charge, the temperature and mois-

ture of the air, must necessarily affect the result somewhat; therefore the foreman first notes the number of revolutions up to the point when the spectroscope tells him that all the carbon has been burnt, and compares it with the estimated number. The difference, if any, is then applied to the calculated number of extra revolutions required for removing the phosphorus, and it depends upon the correctness of these several determinations whether the metal is pure or not. Even the temperature of the charge influences them, for the colder it is the more difficult is it to judge of the disappearance of the carbon lines; so that, generally speaking, it is very easy to be mistaken as to the time when the blast should stop. A danger to which the charge is also exposed is that due to the action of the highly phosphoric slag on the added spiegeleisen and ferro-manganese. It has been confidently stated, but not proved, that the affinity of carbon for oxygen, and of phosphorus for iron, is so great that, unless the latter stage of the process is hurried, phosphorus will leave the slag and re-enter the metal. This seems to be the reason why some works put the admixtures into the ladle and not into the converter. Where this is done the question arises whether the charge can then be thoroughly mixed. If not, this would explain occasional irregularities in the finished product, and these being more apparent in large plates than in bars and wires, may account for the strong dislike entertained towards basic Bessemer plates.

The Acid Bessemer Process differs from the one just described in so far as the lining of the converter is acid (silicic acid = ganister) and that no lime is added. The slag is generated by the combustion of the silicon (to silicic acid) and the iron (to iron oxide). Not a trace of phosphorus is removed by this process, and therefore the pig iron used must contain less than that to be allowed in the finished product. It must also contain a large percentage (2–3 %) of silicon for the production of sufficient heat.

The addition of spiegeleisen and ferro-manganese is effected just before casting, either in the converter or in the ladle.

The spectroscope enables the operator to judge, with reasonable accuracy, as to the percentage of carbon remaining. He could, therefore, either interrupt the blowing before all the carbon is burnt, which would save some of the costly admixtures, or he could wait till all the carbon has disappeared, and then reintroduce it with the necessary manganese. The latter of these two methods is the more reliable. An incidental difference between these processes is that in the basic process, during the necessary after-blow for removing the phosphorus every trace of carbon disappears, while in the acid process this is not the case, so that by adding just sufficient ferro-manganese to remove all redshortness one would obtain a far weaker but also a more ductile material by the basic than by the acid process; the former is therefore used almost exclusively for producing the steel for soft wire.

The Open-Hearth Process was made available for producing mild steel only after Dr. Siemens had invented the regenerative chamber, and although the various shapes of the furnaces take the names of their respective designers, his name will always be associated with steel made by this process.

Heat regenerators consist of several vaults filled with loosely-packed firebricks. Air is admitted through one of these chambers,

and gas through another, and both are ignited in the furnace. The waste products are led to the chimney through the two other chambers, and heat them. The current is then reversed, and the air and the gas pass over the white-hot bricks of the latter chambers, and the waste products heat the former. The process is then continually repeated. Not only does this save nearly all the heat, which would have escaped with the hot gases, but the temperature of the furnace can easily be raised so high that even the firebrick lining would melt.

Within certain limits the temperatures can be regulated by admitting more or less excess air, and by altering the frequency of reversing the current. The flame might also be altered from an oxidising to a reducing one, but not without raising the temperature beyond the endurance of any firebrick. The gas used for firing these furnaces is produced at the works either by partly burning and partly distilling coal, in which case it consists of hydrocarbons, carbonic oxide, and nitrogen; or water gas is used, which contains carbonic oxide, hydrogen, and a smaller percentage of nitrogen. Recently part of the heat in the waste products has been used to distil the coal.

The Acid Siemens-Martin Process.—Pig iron is placed in the furnace, and when melted, iron ore and about 25 % scrap iron are added, until all the carbon and silicon are consumed, and the ore reduced to iron. Samples are repeatedly taken and tested mechanically, to judge of the condition of the bath, and when ready, spiegeleisen and ferro-manganese are added, and the charge is run. As in the Bessemer process, some of these additions may be saved by stopping the process of reduction at an early stage, and in some works the carbon is successfully reintroduced by adding it as powdered charcoal, or anthracite placed in the ladle or into the trough leading to it from the furnace.

During the early period of refining the various layers of the bath are of very different composition, as can easily be ascertained by taking a sample from the top or the bottom of the charge; but a natural mixing takes place, due to chemical action and to an evolution of gases. However at the final stage, when the bath is nearly but not quite uniform, this action practically ceases. It is revived by the addition of iron ores, or of pig, or in some works by stirring with wooden poles, which evolve large quantities of gas.

The acid Siemens process does not remove either phosphorus or sulphur, and the pig and scrap used should therefore contain only traces of these impurities.

The Basic Siemens Process, using Pure Pig, is almost identical with the above, except that, on account of the use of a basic slag, almost every trace of phosphorus disappears. It is also found that both the carbon and manganese are very energetically attacked by the flame, and after the spiegel has been added, it is difficult to hit the right moment for running the charge. This difficulty is aggravated by the fact that the natural lowest limit of mild steel from these furnaces is below 20 tons, whereas with the acid Siemens furnace it is about 24 tons. When trying to make steel of 27 tons, a small error in the admixtures, or in the time of casting, will have two to three times as much effect in the one case as in the other. However with care the very best material, and of the intended hardness, is obtained.

On account of the costliness of the basic lining this process is

hurried as much as possible, and six or even seven charges are sometimes got out of a furnace in twenty-four hours. Of course this is only possible if the refining process is curtailed, and with this object in view not more than about 20 to 25 % of pig iron is used, while the rest is scrap. A further gain has been attempted by carrying out part of the process in a Bessemer converter where all the carbon and silicon are removed. The final reduction takes place in the basic Siemens furnace, where the last trace of phosphorus is abstracted.

The Basic Siemens Process, using Phosphoric Pig, is carried out at a few works in this country. It is a tedious one, lasting about fifteen hours per charge; this is made up of iron ores, lime, and about 75 to 80 % of pig, too poor in phosphorus for the basic converter and too rich for the acid Siemens process. The phosphoric acid which is generated is not volatile, and does not rise to the surface in bubbles like carbonic acid, and therefore cannot assist in mixing the charge, so that the process would be indefinitely prolonged if special means were not adopted, such as the occasional addition of pig and of iron ores, to produce gases. This is necessary up to the last stage, which, as in all other cases, consists in adding spiegeleisen and ferro-manganese.

The basic slag, which floats on the molten steel, is very thick, generally about 12 to 15 ins., and the furnace gases have little chance of acting on the metal. Few works use this process, and there all attempts to manufacture plates on which the same reliance can be placed as on the acid steel have as yet failed. With care good results could no doubt be obtained, but it seems that the effects of even slight carelessness on the part of the operator lead to bad results, and no tests have yet been devised which will detect them.

The Puddling Process has been dying out fast. In it the pig iron is melted in the presence of iron ores and slags by means of flames, which can be changed from oxidising to reducing ones as occasion arises. The temperatures, being very much lower than in all the previous processes, appear to assist at removing a large percentage of phosphorus and sulphur. Much hard labour and considerable skill are required to obtain good results. The final operation consists in extracting the intermixed slag from the iron, which is done under a blooming hammer.

Crucible Steel, as its name implies, is produced in crucibles. These are filled with carefully weighed quantities of blister steel, pure iron, or scrap steel. It is said to be giving way to Siemens steel even for guns, and as it is a very costly process, and has never been used for boiler plates, it is unnecessary to enter into details, except to mention that when carrying out experiments on steel alloys it is of importance that the right fire-proof material should be used. Thus, when wishing to produce practically pure iron, basic crucibles must be employed; if a good percentage of carbon has to be retained, and for many alloys (manganese, silicon, aluminium), the melting must be done in plumbago, and for some other purposes acid crucibles are best, while for certain compositions the melting has to be carried out in various fire-resisting materials, and the molten metal mixed before casting.

The following books will be found to contain a very full account of the manufacture and properties of mild steel :—Dr. J. Percy, 'Iron and Steel,' London, 1864; M. H. Howe, 'The Metallurgy of Steel,' New York, 1890; Professor Ledebur, 'Handbuch für Eisenhüttenkunde,' Leipzig, 1884; V. Deshayes, 'Classement et Emploi des Aciers,'

Paris, 1880; J. S. Jeans, 'Steel,' &c., 1890; Prof. A. Martens, 'Mitt. Berlin,' 1890, vol. ii.

The Influences of Impurities on the mechanical properties of mild steel.—It may be remarked that absolutely pure iron or steel has not yet been produced and experimented upon, and the following remarks, therefore, only apply to the effects of additions made to average qualities.

Carbon increases the tenacity and tempering qualities (if it does not even create them); it reduces the ductility and weldability and melting temperature.

Phosphorus has ascribed to it the chief blame for coldshortness and general treacherousness, but it seems that this is only true if much carbon or sulphur is present. It also increases the liability to get burnt. It reduces the melting temperature.

Sulphur accentuates the bad effects of phosphorus, produces redshortness and greasiness as regards welding.

Arsenic increases the tenacity, reduces elongation and tempering qualities, increases coldshortness as well as redshortness, but only slightly. In chemical analysis it is often mistaken for phosphorus. Like phosphorus and carbon it reduces the melting temperature.

Silicon reduces the elongation and melting temperature, prevents blowholes, increases tenacity and red- and cold-shortness, but only in presence of carbon. It does not accentuate the evil effects of phosphorus.

Tin.—Authorities are conflicting.

Aluminium prevents blowholes, reduces the melting temperature, and hardens cast iron.

Manganese intensifies the influence of carbon, except as regards tempering properties, and neutralises the red- and cold-shortness of phosphorus, sulphur, &c.

Nickel reduces tenacity and increases ductility, particularly as reregards impact. It neutralises the influence of carbon, and perhaps of phosphorus, and it reduces corrosion.

Chromium, like **Tungsten,** intensifies all the influences of carbon, particularly as regards the tempering qualities.

Copper increases tenacity, reduces elongation, and produces redshortness. Its influence on welding is doubtful.

An attempt to summarise these remarks is made in the following table:—

	Carbon	Silicon	Arsenic	Phosph.	Sulph.	Copper	Mang.	Nickel	Chrom.
Tenacity	+	+	+	+	0	+	+	–	+
Elongation	–	–	–	–	0	–	+	+	–
Resistance to impact	–	–	–	–	...	...	+	+	+
Pliability—cold	–	...	–	–	0	...	+	–	–
" dull red	...	–	...	0	–	–	+	+	...
" white hot	...	–?	0	–	0	–	+	...	...
" tempered	+	–	–	...	...	...	–	–	+
Weldability	–	0	...	...	–	0?	0	...	...
Melting temperature	–	...	–	–	–	...	...	...	+
Corrosion	+?	...	...	...	...	...	+?	–	...

NOTE.—0 means that the property is not changed, + that it is increased, – that it is diminished by the impurity; ? means that the authorities are very conflicting; ... means that no information could be obtained.

More detailed information, particularly as to the amount of changes which these various impurities may produce, can be gained by consulting the following papers; but unfortunately where the information is precise the authorities are often in conflict, while generally their experimental results are exceedingly vague. This is due to the difficulty experienced in obtaining pure materials, to the accentuating and neutralising effects of the various impurities on each other, to the neglect and difficulty of analysing the occluded gases, and to ignorance as to the effects of casting temperatures and the various preliminary mechanical treatments.

Carbon.—T. E. Vickers, 'M. E.,' 1861, p. 158; Bauschinger; 'Mitt. Munich'; V. Deshayes, 'An. Mines,' 1879, 7th ser. vol. xv. p. 342; R. H. Thurston, 'Materials,' 1883, vol. ii. p. 420; P. G. Salmon, 'Am. Min. E.,' 1886, vol. xiv. p. 127; H. H. Campbell, 'Am. Min. E.,' 1886, vol. xiv. p. 358. (See p. 60.)

Silicon.—F. Gautier, 'Soc. I. Min.,' 2nd ser. vol. iv. p. 383; H. J. Mräzek, 'Berg- u. H.-J.,' 1872, vol. xx. p. 406; W. Hackney, 'C. E.,' 1875, vol. xlii. p. 35; H. Bell, 'Chem. N.,' 1879, vol. xl. p. 102; H. Bessemer, 'I. and S. I.,' 1877, i. p. 82; Dr. F. C. G. Müller, 'Stahl and Eisen,' 1888, vol. viii. p. 375; ibid. 'Glasers An.,' vol. x. p. 210; P. G. Salmon, 'Am. Min. E.,' 1886, vol. xiv. p. 126; T. Turner, 'Chem. S.,' 1817, vol. li. p. 129; Prof. Tilden, 'Brit. Assoc.,' 1888, p. 69; K. A. Hadfield, 'I. and S. I.,' 1889, ii. p. 222.

Arsenic.—F. W. Harbard and A. E. Tucker, 'I. and S. I.,' 1888, p. 183; Dr. Thomson, 'I. and S. I.,' 1888, p. 133.

Phosphorus.—Prof. Gruner, 'An. Mines,' 1870, vol. xvii. p. 346; M. Euverte, 'Ing. Civ.,' 1874; ibid. 'Soc. I. Min.,' 1876, p. 678; A. L. Halley, 'C. E.,' 1878, vol. liii. p. 221; Dr. F. C. Müller, 'Deut. Ing.,' 1878, vol. xxii. p. 385; ibid. 'Mon. Ind.,' 1881, Sept. 15; Dr. Thomson, 'I. and S. I.,' 1888, p. 193.

Sulphur is mentioned by nearly all experimenters.

Aluminium.—R. A. Hadfield, 'I. and S. I.,' gives a full list of previous experimenters.

Copper.—A. Wassum, 'Soc. I. Min.,' 1882, vol. ii. p. 192; ibid. 'Chem. Soc.,' 1883, p. 404; M. Euverte, 'Soc. I. Min.,' 1884, p. 85; Dr. E. J. Ball, A. Wingham, 'I. and S. I.,' 1889, i. p. 123.

Manganese.—M. Caron, 'Comp. Rend.,' 1863, vol. lvi. p. 828; F. Gautier, 'Soc. I. Min.,' 2nd ser. vol. iv. p. 383; M. Euverte, 'Ing. Civ.,' 1874; ibid. 'Soc. I. Mines,' 1876, p. 678; Åkermann, 'Eng. Min. J.,' 1875, ii. p. 214: J. Riley, 'I. and S. I.,' 1877, i. p. 195; Dr. H. Wedding, 'Ver. Gew.,' 1881, p. 509; H. H. Campbell, 'Am. Min. E.,' 1886, vol. xiv. p. 358; T. Turner, 'Chem. Soc.,' 1886, vol. li. p. 138; R. A. Hadfield, 'C. E.,' 1888, vol. xciii. p. 1. (See p. 50.)

Nickel.—J. Riley, 'I. and S. I.,' 1889, i. p. 45.

Mechanical Tests.—Besides chemical impurities there exists another set of influences which affect the qualities of steel and iron, but as most of them are only apparent when testing the material, their discussion will be deferred until such of the various tests have been described as have been suggested or adopted for the purpose of ascertaining the quality of steel and iron. In Germany these tests are called 'Qualitätsproben' (quality tests), which expresses very appropriately that they are mainly intended to ascertain the quality, and not,

as is too often assumed in this country, that they should be an imitation of the conditions to which the material is to be subjected in the structure. Looked at in this light there is nothing unreasonable in bending a sample which has been cut from a plate which will never be subject to bending stresses, or to test a sample by tension, when the plate will certainly only be subjected to compression stresses. The object of the various tests is to ascertain the quality, and that is all, the correlation between a test and the quality best suited for a particular purpose having been determined by practical experience.

Thus, when iron came into use it was found that the better qualities—those which did not break with ordinary usage—could under the conditions of testing stand bending through a certain angle, and when fractured showed certain peculiarities of grain called fibre. Tests on this line, of a more or less severe nature, were then looked upon as a criterion of the quality. Later on tensile tests were added. In the early days of steel it was found that these tests did not suffice; that plates cracked without apparent cause when in use, even though samples which had been cut from them had withstood all the tests prescribed for the best iron in a more than satisfactory degree. It was found that this material was not reliable unless it would pass the temper-bending test, which was then adopted. The cold bending test was dropped, but since basic steel is being produced it has been found necessary to revert to it, and above all to carry it out without filing or machining the sheared edges.

The following are the most important of these mechanical quality tests.

Cold Bending.—This test must be looked upon as a very important one, and more certain to detect a defective plate than either the temper-bending or tensile tests. To punch holes in the samples before bending does not appear to give a better indication as to the natural quality of the steel than if the sample is bent with sheared edges; but care should be taken that the shearing is done from opposite sides of the plate, and the two sharp edges should be kept on the inside of the bend; otherwise the results will be very erratic and unsatisfactory, depending not so much on the injury to the metal which has been caused by the shearing as on the shape of the ragged edge. If a tear commences from the sharp edge, it should not be looked upon as being very serious. There seems to be a difference, depending on whether the concave or convex side of the sample was the upper one while the ingot was heated for rolling, but nothing reliable is known about this subject.

Inferior qualities of steel will not behave well, and occasionally it happens that a strong steel bends better than a weak one. The bending of samples after annealing them is valueless, unless they crack even then, which would be an unmistakable sign that the material is quite bad. The thickness of the plate is an important factor, and when 1 in. is exceeded only the best steel having sheared edges will stand even the slightest bending. For $\frac{3}{4}$-in. plates and upwards it is well to plane off $\frac{1}{8}$ in. of the edges, but then the samples should bend quite double. Care must be taken that the temperature is a reasonable one—say, 82° F. or 100° F.—for excessive cold has a bad influence, as will be seen from the following experiments. (See also p. 112.)

Out of ten samples cut from one $\frac{3}{8}$-in. plate six bent double at 82° F., while the four others broke at angles varying from 90° to 100°. These had been bent at 0° F. (i.e. 32° below freezing).

Two sets of samples of $\frac{1}{2}$-in. plates were cut from six separate charges. The first three were bent at 82° F. to $\frac{3}{8}$-in. radius, while at 0° F. the duplicate samples broke at $1\frac{1}{2}$-in. radius. Three others were bent at 82° F. and broke at 1-in. radius; their duplicates, tested at 0° F., broke off short. There was nothing in the analysis to indicate that the two qualities were different.

This cold bending test is the only one which seems likely to detect whether the steel possesses the disagreeable quality of reverting to its (perhaps) naturally bad condition. When this is feared duplicate samples with sheared edges should be kept for a week or more, bent, and their curvatures compared with the original ones. Both samples should be bent till they crack, and at identical temperatures.

The Temper-Bending Test was devised to guard against the employment of steel liable to crack after it had been worked. Combined with the condition that the tenacity might reach thirty-two tons, it compelled manufacturers to replace part of the carbon in the steel by manganese, thereby maintaining the tenacity, but reducing the tempering qualities (see p. 105). The test is carried out as follows: Steel samples are heated to a cherry-red, and suddenly cooled in water of 82° F. (=28° C.) When cold they are bent. Better bending results are always obtained if a large batch of samples is operated upon, for if they are all thrown into the tank together their mass sufficiently retards the cooling and hardening effect of the water, while if thrown in singly none except the first few are exposed to the very cold water. Should there be any doubt about the quality of particular charges, it would not be difficult to retain such samples to the last, when the water has grown warm.

Another means of improving the bending results is to take the samples out of the water trough while still warm, and bend them in this condition. The effect of quenching the samples from a dull red heat seems to be a bad one, but the reverse is true if they are annealed thoroughly at a good red heat for a quarter of an hour, and then cooled slowly to a cherry-red before quenching. The use of a bending press instead of a hammer does not seem to be an advantage; in fact, experience points the other way, and an explanation may be sought for in the fact that most bending presses set up a greater tension stress in the outer fibres than is the case with a steam hammer. The removal of sheared edges before tempering—at least as regards the best qualities of steel—seems to be injurious. J. Riley arrived at a different conclusion ('I. and S. I.,' 1887, p. 121).

The Red-hot Bending Test is sometimes carried out on plates to be flanged, but seems to be unnecessary, except for iron. With steel this or a similar test is carried out on the baby ingot which is cast during the running of the charge, so as to be sure that the steel will roll well.

For mild steel plates the following test is occasionally adopted :—

While red hot, a half-inch hole is punched in a strip of metal about $1\frac{1}{2}$ in. wide (see fig. 112), and is then drifted to 1 in. diameter and the end slit open, as shown in fig. 113; the two arms are then bent

as shown in fig. 114, and the drifted surface should show no signs of cracks. This work is done in two heats.

The Drift Test consists in forcing a cylindrical cone into a drilled hole alternately from one side and then from the other. It is some-

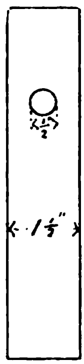


FIG. 112.

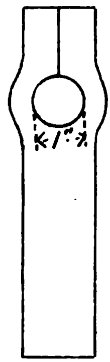


FIG. 113.

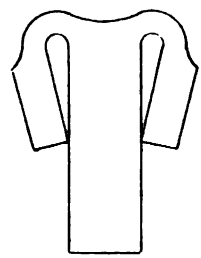


FIG. 114.

times applied to steel intended for tube plates. It is a tedious and very uncertain test, and gives no indication of the quality of the material. This was very conclusively shown in W. Hackney's paper on test pieces ('C. E.,' 1883, vol. lxxv. p. 70). Locomotive tube plates are sometimes tested in this way.

Alternate Bending Tests are very tedious, and have to be carried out with much care, but then they are most valuable, as an indication of the ductility of the material. These tests are much used for wire, and they have given good comparative results in the author's investigations on blue heat. The plan there adopted was to bend the samples alternately from one side to the other through an angle of 45° over a radius of $1\frac{1}{2}$ in. Professor Wöhler ('Zeitschrift für Bauwesen,' 1860, vol. x. p. 583, and subsequent volumes), and recently other experimenters, have subjected samples to repeated alternate stresses, only slightly exceeding the elastic limit of the material. This was done in order to ascertain the amount of *fatigue* which a material can withstand.

Bending Tests with Nicked Samples.—This test is a very valuable one if applied to iron, as it exposes the grain. With good steel the result is unsatisfactory, because it is difficult, even after nicking, to break it. In order to obtain a fracture the sample should be nicked all round, or bent through an angle of about 30° before nicking, and then broken.

Fractures.—The shape of a fracture depends not only on the quality of the material, but also on the method of breaking; if done slowly fibre usually appears. Sudden blows produce crystals; their size increases with the softness of the material. It is a mistake to imagine that these crystals exist in the material; they are the shape of the fractured surface. The starting-point of most fractures can easily be found.

Percussion Bending Test is customary with rails and with tyres. The comparatively thin plates of a boiler are nearly always subjected to it as they are struck with a hammer, and, as already mentioned, the

results are, if possible, more satisfactory than when the bending is done in a press.

Tensile Tests.—Much importance is attached to this test. The ease with which it can be explained and the uniformity now obtained are powerful inducements for retaining it. It is useful for making comparisons of quality independent of tenacity, though in this respect it is not nearly as valuable as some of the bending tests; they, however, require observers possessed of much experience. The primary object of the tensile test is to ascertain the ultimate strength of the material; but the permanent elongations also give indications of the ductility, while with accurate instruments the modulus and limit of elasticity can also be ascertained. No relation between the contraction of area and general qualities has been detected, except that basic steel contracts more with equal elongation than acid steel. The temperature has a marked influence (Dr. J. Kollmann, 'Ver. Gew.,' 1880, vol. lix.).

A good deal has been written on the forms of test pieces, and the influence of length, breadth, and thickness, and on the presence and absence of scale on the surface—E. J. Reed, 1869, p. 401; P. D. Bennett, 'M. E.,' 1886, p. 27; E. Richards 'I. and S. I.,' 1882, p. 11; Dr. H. Zimmermann, 'C. E.,' 1883, vol. lxxiv. p. 301; W. Hackney, 'C. E.,' 1883, vol. lxxv. pp. 70–159; D. Kirkaldy (various); J. Bauschinger, 'Mitt. Munich,' 1892, vol. xxi.

The general conclusions are, that long, thin test pieces show less strength and do not elongate nearly as much as short ones, but this is only true for materials which contract at the point of fracture. Mild steel contracts very much, so that results as to elongation would be misleading unless the length of the sample is mentioned. The general practices as to length are—

Board of Trade, 10 ins.
Admiralty, Lloyd's, and Continent, 8 ins., or 200 mm.
Artillery experts, Whitworth, and railway tyres } about 2 ins.

This short length of 2 ins. is due to the want of dimensions in the gun barrels. The removal of the scale reduces the elongation and increases the tenacity.

The Testing Machines all work on the principle that the sample is attached by one end to a weighted lever, and by the other end either to a screw or press which can be adjusted to make up for the elongation. With the older machines the screws can only be worked after the load has been removed, and the testing of ductile materials cannot be carried out in one operation, but this does not seem to affect the results.

In some machines the samples are placed horizontally, in others vertically, and the levers are either simple or compound, and the loading is done either by a jockey weight travelling on a lever, by weights added to the extremity of the lever, or by a fluid pressure or a pendulum weight.

The oldest machines are worked by added weights. On the Continent (chiefly Austria) water is run into a large bucket at the end of the lever. Generally a weighing machine is permanently stationed under the bucket, so that it can easily be weighed, but a gauge glass is also attached.

In England single levers and jockey weights are now customary. The stretch is taken up by steam power acting on a ram, and the speed is about 3 inches stretch per minute.

At some German works the lever is a loaded pendulum, which rises while a hydraulic ram stretches the sample.

In France and Belgium the lever is a crank, which presses on an iron disc resting on a sheet of leather, which covers a basin filled with mercury, and which communicates with a graduated vertical glass tube. The total load can be read off on a scale. This arrangement gives very accurate results, and can with advantage be used for ascertaining the drop (or elastic limit); but it is unsatisfactory in cases of dispute, because the column of mercury sinks back to zero at once when the test piece is broken. The defect could be remedied, or by attaching an ordinary steam indicator automatic diagrams could be obtained.

When the elastic limit has to be ascertained strain indicators are indispensable. A description of these will be found in 'C. E.,' 1887, vol. lxxxviii. p. 1.

Descriptions and illustrations of testing machines will be found in the following publications: J. H. Wickstead, 'M. E.,' 1882, p. 384, and 1886, p. 27; Prof. A. B. W. Kennedy, 'C. E.,' 1887, vol. lxxxviii. p. 1; U. R. Towne, 'M. E.,' 1888, pp. 206, 448. 'Engineering,' vol. xxxi. p. 57; vol. xxxiv. p. 254; vol. xxxv. p. 346; vol. xxxvi. p. 146; vol. xli. p. 180; vol. xliii. pp. 414, 572; vol. xliv. pp. 649, 652; vol. xlv. p. 458, &c.; vol. xlvi. p. 21.

Many testing machines are so arranged that they can be used for measuring compression-bending and torsion stresses. Appliances for bending samples can be attached to almost any machine, but that is not the case as regards compression and torsion. In the former case very careful adjustments are necessary, in order that the thrust may be perfectly central and the bearing perfectly normal. In both respects most machines are very imperfect. Almost any lathe can be used for the torsion test. It is a valuable one, but not carried out often enough, and then the lessons it teaches are not properly understood (see p. 125).

Percussive Tensile Tests are carried out by securing the top end of a tensile test piece to a strong but narrow beam, and attaching a cross-bar to its lower end. A heavy weight, whose lower end is forked, is then dropped on it, striking the cross-bar. Another plan is to attach the weight to the lower end of the test piece and a cross-bar to its top end; the whole is then raised and dropped; the cross-bar is arrested by stops, while the weight tears the sample asunder. This is the more usual plan, and seems to be the better of the two.

Colonel Maitland ('C. E.,' 1887, vol. lxxxix. p. 114) was able to obtain a still more sudden rupture by shaping the test piece somewhat like a dumb-bell, surrounding the central, thinner part with gun cotton, and inserting both into a strong tube open at both ends. On firing the explosive the two ends were driven out with great violence in opposite directions. It was found that during rupture the samples had elongated very much. This, however, might be due to combined action of the longitudinal pull and the surface pressure on the bar, acting like the forces which come into play when drawing wire (see p. 128).

Chipping, Machining, and Scratching have repeatedly been proposed for use, as they seem capable of giving valuable information. Thus the turning tool of a lathe will show up very distinctly the various slabs and layers in an iron bar, exposing, as it were, very slight differences in the hardness or toughness of the material. With a hammer and

chisel there is no difficulty in distinguishing between iron and steel, and some boilersmiths profess to be able to tell, with the help of these tools, where a particular piece of steel has been manufactured. This test has not yet been made practical; the same may be said of *etching* and the *microscope*, and *magnetic tests*. Prof. Egleston, 'Am. M. E.,' 1879, vol. v. p. 140. S. M. Saxby, 'N. A.,' 1884, vol. ix. p. 61; 1885, vol. x. p. 119. Prof. D. E. Hughes, 'M. E.,' 1884, p. 36. C. M. Ryder, 'Metall. Rev.,' 1877–8, vol. i. p. 317. Prof. K. Keller, 'O. I. A. V.,' 1879, vol. xxxi. p. 163. Dr. H. Wedding, 'Stahl und Eisen.' J. A. Ewing, 1891.

With the help of some of the above-mentioned tests it has been attempted to solve the various problems of the behaviour of iron and steel. Some have been cleared up, others remain mysteries.

The Influence of Temperature, particularly of heat, on the strength of materials is a subject of the first importance. The relation existing between the boiler pressures and temperatures will be seen from the following table:—

Boiler Pressure	Lbs.	0	50	100	150	200	250	300	400	500
Temperature	°F.	212	281	328	358	382	401	417	445	467
"	°C.	100	138	153	181	194	205	214	230	242

Original researches on this subject will be found in the following papers:—'Franklin Inst.,' 1836, ii. pp. 82–208; M. Baudrimont, 'An. Ch. Ph.,' 1850, iii. vol. xxx. p. 304; W. Naylor, 'M. E.,' 1866, p. 76; W. Fairbairn, 1856, 2nd ser. p. 96; Sir W. Fairbairn, 'Manch. L. Ph.,' 1871, vol. x. p. 86; 'Portsmouth Dockyard Experiments,' 1877; C. Huston, 'Frankl. Inst.,' 1878, vol. lxxv. p. 93; 'N.,' 'Glasers An.,' 1880, vol. vii. p. 165; Dr. J. Kollmann, 'Ver. Gew.,' 2nd ser., 1880, vol. lix. p. 92; J. F. Barnaby, 1881 and 1882; J. E. Howard (Watertown Arsenal), 'I. and S. I.,' 1889, ii. p. 460; Le Chatelier, 'Comp. Rend.,' 1889, vol. cix. p. 58; Board of Trade Report (No. 257) on 'Boiler Explosions'; Prof. Martens, 'Mitt. Berlin,' 1890, p. 159; A. Bleichenden, 'M. E.,' 1891, p. 320.

The Influence of Cold on the tenacity of metals is dealt with by the following experimenters:—Sir W. Fairbairn, 'Parl. Report of the Commissioners of Railway Structures,' p. 321; ibid. 'Brit. Assoc., 1857, p. 405; K. Styffe, 1869; 'N.,' 'Glaser's An.,' 1880, vol. vii. p. 165; Capt. Bernardo, 'Rev. d'Art.,' 1890, p. 485; W. Brockbank, 'Manch. L. Ph.,' 1871, vol. x. p. 77; W. W. Beaumont, 'C. E.,' 1876, vol. xlvii. p. 43; T. Andrews, 'C. E.,' 1887, vol. lxxxvii. p. 340; Spangenberg, 'Glaser's An.,' 1879, vol. v. p. 165; T. Andrews, 'C. E.,' 1891, vol. cv. pp. 161, 169.

Experiments on the influence of temperature on copper and other materials:—'Frankl. Inst.,' 1836, ii. p. 39; Dr. Kirk, 'Enging.,' 1887, vol. xliv. p. 661; W. Parker, 'N. A.,' 1889, vol. xxx. p. 47; Le Chatelier, 'Comp. Rend.,' 1889, vol. cix. p. 24.

Dr. J. Kollmann's experiments on iron and steel, 150 in number, seem to be the most reliable. His conclusions are that the ultimate strength of both materials decreases with rising temperature, and that it shows a very serious drop at about 450° to 500° F. The elongation is at a maximum at about 900° F. The contraction of the fractured sectional

area steadily increases till it reaches 90 % at a red heat. The limit of elasticity decreases steadily.

Another result is that the ductility, as measured by bending tests, increases with rising temperatures till 450° to 500° F. (blue heat) is reached, when the material is rotten. At higher temperatures it is pliable once more. It would seem as if medium quality steel is exceedingly brittle when cold (0° F.), while tougher qualities of the same tenacity remain ductile; at any rate, boiler-smiths and ship-platers have come to the conclusion that it is risky to handle steel or iron plates in cold weather, and where serious hammering or drifting is contemplated, heaters are almost invariably applied. If the steel contains much phosphorus it is exceedingly likely to break under the percussive bending test, if carried out in cold weather.

Sudden cooling, not necessarily from a red heat, is said to produce brittleness (see T. Andrews, 'C. E.,' 1891, vol. ciii. p. 231); and repeated heatings have also produced brittleness, but the exact conditions for effecting this change are not known. E. Wehrenfennig, 'O. I. A. V.,' 1879, vol. xxxi. p. 153; J. P. Barnaby, 1881 and 1882; A. Ledebur, 1884, p. 646; C. E. Stromeyer, 'C. E.,' 1886, vol. lxxxiv. p. 122; E. B. Martens, 'Ing. Civ.,' 1886, p. 607; E. Wehrenfennig, 'Organ,' 1884, vol. xxi. p. 216; A. E. Sherk, 'Enging.,' vol. xliv. p. 458; C. E. S., ibid. vol. xliv. p. 491; B. H. Thwaite, ibid. vol. xliv. pp. 505, 536; Th. Edington and Son, ibid. vol. xliv. p. 505.

Occlusion of Gases.—In some of the above-mentioned cases the brittleness may have been due to the absorption (occlusion) of injurious gases, and experiments prove that hydrogen is readily absorbed, and that it injures the material. Acids and other corrosive influences are also said to produce brittleness, but the real cause may be the hydrogen which is evolved during these processes.

Dr. Schahäntl ('Bairisches Kunst- und Gewerbeblatt,' June 1863) gives an analysis of the various layers of an exploded boiler plate, and shows that on the water side it contained an excess of oxygen, while on the fire side occluded sulphurous acid was found. That solids do absorb gases is proved by the well-known fact that soaped window glasses turn a mauve colour after a time, which is due to a chemical action of the oxygen of the air on the manganese salts in the glass.

M. Bustein ('An. Mines,' 1883, viii. vol. iii. p. 28) gives the chemical analysis of three qualities of steel which had been exposed for 112 days in flue gases or in boiler water.

Previous Treatment of Sample	Tenacity—Tons			Elongation—per Cent.		
Original	56	50	42	17	19	24
Exposed 112 days in boiler . .	51	47	40	15	16	14
" " flue . .	44	41	37	16	21	"

Prof. Hughes ('Tel. Eng.,' 1880) deals with this subject.

Influence of Pickling.—A. Ledebur ('Stahl und Eisen,' 1887, vol. vii. p. 682) gives analyses of eight qualities of steel (wire), and finds that pickling reduces the elongation about 15 %, and the ductility 39 %; that exposure to the atmosphere for two months reduces both qualities about 50 %, but that annealing puts matters right again, though the ductility is not perfectly restored. The action of zinc in galvanic con-

tact with the wires is worthy of notice; it prevented corrosion, but the wires grew very brittle, and on analysing them again they were found to contain from ·002 to ·005 % of hydrogen. He points out that, on account of the low atomic weight of hydrogen, these percentages should be multiplied by, say, thirty to make them comparable with the volumes of phosphorus or of sulphur. Baedecker ('Deut. Ing.,' 1887, vol. xxxii. p. 187) confirms these views about the action of acids.

The study of this subject is as yet in its infancy, and is certainly beset with many difficulties. It may, therefore, be of interest to mention where experiments on occluded gases can be found:—L. Troost and P. Hautefeuille, 'Comp. Rend.,' 1875, vol. lxxx. p. 788, and 'An. Chim. Ph.,' 1876, 5th ser. vol. vii. p. 155; M. Reynard, 'Ing. Civ.,' 1877, pp. 91 and 210; A. H. Allen, 'I. and S. I.,' 1879, p. 480, and 1880, p. 181; F. C. G. Müller, 'Deut. Ch. G.,' vol. xii. p. 11; ibid. 'Glaser's An.,' 1880, vol. vii. p. 138; ibid. 'Stahl und Eisen,' 1882, vol. ii. p. 531; N. Zyronski, 'Soc. I. Min.,' 1884, p. 101; H. M. How, 'Eng. Min. J.,' vol. xlv. p. 236.

It would be wrong to compare occlusion by metals with the absorbing power of fluids—for instance, with water, which takes up very much larger quantities of carbonic acid when cold than when warm. On the contrary, metals behave in a most erratic manner, and usually absorb certain gases in a molten state, which are given off again when cooling. Thus molten silver and copper absorb oxygen, and expel it with much force as they harden. Therefore it can hardly be expected that the annealing process will drive out all the gases occluded by iron or steel, even if carried out in a vacuum, although a certain proportion would disappear. The gas given off by steel in the soaking pits is said to be hydrogen.

Burnt Iron.—It has been attempted to explain the mystery of burnt iron as being due to the presence of a large percentage of occluded oxygen; but iron can unquestionably also be 'burnt' in a vacuum and in gases containing no oxygen. This has led to the view that the occluded gases in the iron or steel—chiefly hydrogen and nitrogen—leave the metal and form innumerable cavities, thereby destroying its continuity and making it rotten (burnt).

W. M. Williams ('Chem. S.,' 1870, vol. xxiv. p. 790) found oxide of iron in burnt iron.

H. Caron, 'Comp. Rend.,' 1872, vol. lxxiv. p. 662. Iron could be burnt in air, hydrogen or nitrogen gas.

Professor Ledebur, 'Jahrb. B. H.,' 1883, p. 19. Phosphoric iron is more easily burnt than pure qualities. He believes burning not to be due to oxygen, and mentions **dead iron.**

In spite of this diversity of opinion there can be no doubt that iron and steel do get burnt if exposed to excessive heat, and are then both red- and cold-short; and also that in re-heating furnaces, and even in annealing furnaces, burning can very easily be brought about if the fire grates are left partly uncovered and pure but heated air is allowed to impinge on the plates.

Recalescence.—It might be that burning is simply due to an excessively high temperature, for iron perhaps and steel certainly exhibit strange phenomena when heated. One of these has been called recalescence, to which attention was first drawn by Mr. Barret, who showed

that if a piece of hard steel is heated to redness and then allowed to cool slowly it will, when almost black (in a dark room), suddenly reglow or recalesce, and then get dark again.

The subject has been very exhaustively investigated, and seems to be closely related to the specific heat of iron, which shows strange irregularities at certain temperatures, occasionally changing from a positive to a negative quantity, i.e. the temperature rises while heat is being abstracted.

Barret, 'Phil. Mag.,' 1873, vol. xlvi. p. 472.

G. Forbes, 'R. Soc. Edinb.,' 1873-4, vol. viii. p. 363.

M. Pionchon, 'Comp. Rend.,' 1886, vol. cii. p. 1451. Specific heats of iron up to 1,000° C. ; breaks at 660° C. and 723° C.

F. Osmond, 'Comp. Rend.,' 1887, vol. ciii. pp. 743, 1135. Recalescence of mild steel between 670° and 690° C., 735° and 775° C., 820° and 860° C., and other periods for harder steels.

Brinell, 'I. and S. I.,' 1886, p. 365.

A. Ledebur, 'Stahl und Eisen,' 1887, vol. vii. p. 447. Recalescence of four qualities of steel and two of iron (analysis). There is a difference between cooling and heating the samples.

H. Tomlinson ('Phys. S.,' vol. xix. p. 107) notices seven points of recalescence.

H. F. Newall, 'Phil Mag.,' 1888, vol. xxv. p. 510.

F. Osmond, 'I. and S. I.' 1890, I. p. 38. Recalescence of fifteen qualities of iron and steel. Pure (electrolytic) iron possesses the recalescent properties, but ferro-manganese does not.

Temnikoff, 'Gorni J.,' 1887, p. 308.

T. Andrews ('C. E.,' 1888, vol. xciv. p. 192) gives temperature readings down to 0° F., which show two distinct breaks similar to those noticed by F. Osmond and others at high temperatures.

The above experiments show that the recalescent property is possessed by pure iron, and that chemical admixtures modify it somewhat. It is worthy of note that a large percentage of manganese removes the property, and also makes steel very ductile at a blue heat, which is a temperature corresponding with one of the recalescent periods.

Other metals have not been experimented upon in this line, but the task might be a profitable one, particularly as regards copper, which is exceedingly brittle at a dull red heat.

A careful study of the above might also throw light on the question of annealing and tempering (or quenching) ; for it is strange that copper, brass, and manganese steel should be hardened by annealing and softened by quenching, while steel behaves exactly in the reverse way. It is evident that the chemical composition, particularly the carbon, plays an important part, and it even seems as if absolutely pure iron hardens if annealed.

Annealing.—It is well known that annealed iron wire has an initial hardness, which disappears on straining it. Basic (Siemens) steel, made of pure pig and low in carbon, usually shows a slight increase in its ultimate tenacity after annealing, while tempering lowers its elastic limit. The reverse is the case with acid Siemens steel, which generally contains more carbon and less manganese. Internal strains are never entirely removed by annealing (p. 23). Tool steel can be thoroughly annealed by heating it to redness and cooling slowly till it is just

dark, and then quenching. Mild steel gives better bending tests if quenched at a cherry-red heat than if quenched at a very dull red heat. The literature on the subject is chiefly restricted to the question of removing the injury done by punching, and to the influence of quenching on the comparatively hard steel used for guns.

Effects of Quenching Red-hot Steel.—For thin plates it seems advantageous to remove the sheared edges before tempering the samples, while for thick plates the reverse appears to be true. Quenching in oil gives more toughness, while quenching in water, and particularly in mercury, increases the hardness. Quenching at a dull red heat thoroughly anneals tool steel. Quenching in molten lead or in boiling water produces nearly the same effect as annealing. This should not be done to any structures exposed to corrosive influences, as the minute particles of lead which adhere to the iron produce very severe local corrosion.

Galvanised Steel. Injury, possibly due to chemical action or to the absorption of vapours, is produced if iron or steel articles are galvanised at too high a temperature. In order to guard against this danger, the fires for heating the zinc baths should always be placed round their sides ; this allows the impure zinc (containing iron) to fall to the bottom, where it floats on a layer of molten lead. If mixed with the other zinc the temperature of the bath would have to be raised too high.

Another class of influences —chiefly mechanical treatments—have to be noticed, as they affect the quality of iron and steel.

The Influence of Hammering or Cogging the Cast Ingots, or of Rolling them direct.—In England the practice is either to hammer or cog the ingots, while in Germany this is not always done, which is made possible by the different chemical compositions. Hammered ingots show a better surface when rolled, and are said to be a little denser. The question of waste is intimately bound up with the above, and there can be no doubt that there is less scrap from plates whose ingots were cogged or hammered than if no preliminary shaping had taken place. From a boiler-maker's point of view a large amount of scrap is an advantage, for the edges are sometimes overheated, and are never of exactly the same quality as the rest of the plate. As regards the final quality of the material several important investigations have been carried out, but being in each case confined to steel of one company, the deductions are not conclusive as regards other makers.

J. Riley, 'I. and S. I.,' 1887, p. 121. Influence of rolling, hammering, and annealing. This is a very exhaustive paper, and the conclusion to be drawn from it is that an excessive amount of hammering and rolling is not necessary. From one and the same charge thick plates are both weaker and more ductile than thin ones, which have received more work.

W. Parker ('I. and S. I.,' 1887, p. 134) mentions experiments to show that tenacity and elongation increase with rolling.

H. Allen, 'C. E.,' 1888, vol. xciv. p. 240. Rolled wire and drawn wire are neither much stronger nor more ductile than the billet unless tested in an unannealed condition.

D. Kirkaldy gives a few experiments.

W. H. Greenwood, 'C. E.,' 1889, vol. xcviii. p. 83. Fluid compressed steel. The benefit derived from this preliminary compression does not seem to dissipate during the subsequent manipulations.

The immediate effects of hammering, &c., are investigated in the following papers :—

H. Tresca, 'M. E.,' 1878, p. 315.

Ibid., 'Comp. Rend.,' 1883, vol. xcvii. pp. 222, 515, 928. Localisation of heat developed by a hammer blow.

M. Lau, 'Comp. Rend.,' 1882, vol. xciv. p. 952.

M. Osmond, 'Comp. Rend.,' 1885, vol. c. p. 1228. Motive power for rolling steel is $2\frac{1}{2}$ times as much as for iron.

Dr. J. Kollmann, 'Ver. Gew.,' 1880, 2nd ser. vol. lix. p. 6.

Local Heating.—Serious injury is sometimes done to a steel plate by drawing out one of its corners, even though this be done at a red heat. Flanging, whether done by hand or presses, has also led to failures. It has never yet been reasonably demonstrated that this is due, as some contend, to the local heating alone, or, as others believe, to the straining produced by the unequal expansion of the locally heated parts ; other influences, such as the chemical composition of the material and the processes through which it has passed, are probably important factors.

The following list contains published cases of cracked plates :—

Sir N. Barnaby, 'I. and S. I.,' 1879, p. 242. List of forty three steel failures at Chatham in six months.

W. Denny, 'N. A.,' 1880, vol. xxi. p. 185. List of steel failures in his own yard.

W. Parker, 'N. A.,' 1881, vol. xxii. p. 12. Cracked plates of the steam yacht 'Livadia.'

A. C. Kirk, 'N. A.,' 1882, vol. xxiii. pp. 131 and 137. Cracked flanged plate.

J. F. Barnaby, 'Enging.,' 1883, April 20 ; D. S. Smart, 'C. E.,' 1884, vol. lxxx. p. 102. Sketches of cracked plates.

W. Parker, 'N. A.,' 1885, vol. xxvi. p. 253. Failures of thick steel plates.

H. Goodall, 'C. E.,' 1888, vol. xcii. p. 10. Sketches of some cracked plates. (See also p. 199).

Of course this list contains only a very small fraction of the number of cracked plates, for as long as it is not settled whether these failures are due to the material, or to the mode of working the plates, it will be the custom, as now, for the steel-makers to replace the failed plates.

Blue Heat.—Perhaps some, though certainly not all, such failures are due to working the steel at a blue heat. This is another mysterious phenomenon connected with iron and steel. The salient features are that steel will bend without fracture at temperatures ranging from 0° F. to 450° F., and again above 550° F., but it is quite rotten between these limits ; and, further, a piece of steel which has been bent or hammered at this particular temperature, but without breaking, acquired and retained a permanent excessive brittleness (see p. 203). This brittleness can be removed by annealing, but time alone (8 years) does not restore the quality. Experiments on the subject, as well as failures, which are attributed to working steel at a blue heat, will be found in the following papers :—

M. Valton, 'Berg- H.-Z.,' 1877, vol. xxxi. p. 25 ; D. Adamson, 'I. and S. I.,' 1878, p. 402 ; W. Denny, 'N. A.,' 1880, vol. xxi. p. 185 ; C. E. Stromeyer, 'C. E.,' 1886, vol. lxxxiv. p. 114 ; ibid., 1888, vol. xciii. p. 89 ; W. Parker, 'I. and S. I.,' 1887, p. 136 ; G. B. Craig,

'N. A.,' 1888, vol. xxix. p. 113; Rudelloff, 'Mitt. Berlin,' 1889, p. 97; Board of Trade Report, August 31, 1886; 'Am. R. M. M. A.,' 1892; Prof. A. Ledebur, 'Glaser's An.,' 1886, vol. xviii. p. 205.

Besides these cases a few illustrations will be found on p. 199, and it is also possible that some of the failures noted in the following two lists may be due to the same cause, or perhaps to the

Influence of Time.—There are indications that exposure or time alone can change a tough material into a brittle one in the same way that elastic (amorphous) sulphur slowly changes into the hard and brittle condition. It is also said that the nature of pure tin, as well as of nickel steel, is permanently changed when exposed to cold.

List of Spontaneous Failures.—Z. Colburn, 1860, p. 32. Old boiler stays said to be brittle.

Professor Thurston, 'I. and S. I.,' 1875, p. 342. Old rails had grown brittle, and improved on re-rolling.

Ibid., 'Materials,' 1883, vol. ii. p. 576. Prolonged excessive straining causes rupture.

Collingwood, 'Am. C. E.,' 1880, vol. ix. p. 171. Tenacity of wire changes after a time.

L. Fletcher, 'C. E.,' 1884, vol. lxxx. p. 136.

A. J. Maginnis, 'Engr.,' 1885, vol. lx. p. 447; C. E. Stromeyer, 'C. E.,' 1886, vol. lxxxiv. p. 187. Sketches of plates which cracked spontaneously while not in use.

J. Harrison, 'Engr.,' 1886, vol. lxii.

'Army N. J.,' 1887, vol. xxiv. p. 65. A long list of steel armour plates which cracked spontaneously before being fitted.

Collingwood, 'Am. C. E.,' 1880, vol. ix. p. 171, and W. Hewitt and Felton, 'Am. M. E.,' 1888, vol. ix. p. 47. Strength and ductility of wires and plates change 24 hours after rolling.

H. M. Howe, 1890, p. 195, old armour plates are brittle; p. 210, time removes injury caused by cold working.

For other cases, see p. 23.

Although not as yet published, there are a few cases of boiler shells cracking under the hydraulic test, and particularly of furnace saddle corners cracking while out of use or if struck with a hammer. One of the most curious cases is perhaps the following: A steel steamer which had been on fire was being repaired, and several of the buckled plates had been taken out to be straightened. It was found that some of these cracked spontaneously while lying on the ground, although they had not cracked while being removed from the ship.

Besides these various failures other instances will readily suggest themselves about the influence which time has on the quality of iron and steel. There is the case of the gun of H.M.S. 'Collingwood,' which burst with a light charge after two years' rest, succeeding on its trial with several proof charges. These and similar failures are so very mysterious that they have been attributed to hidden or incipient flaws; but they have never been seen, and if the point is conceded that certain qualities of steel have a natural tendency to change—and this is stoutly maintained by experienced steel manufacturers—then the difficulty is somewhat reduced. That important changes are slowly occurring in steel is proved by the fact that ordinary test samples which have been successfully bent to a small radius crack spontaneously some time

afterwards; that armour plates, after having been fired at, emit strange sounds for a long period, and that the elastic limit of steel test pieces which have been stretched a few per cent. slowly grows higher and higher when at rest.

Time can hardly be called a treatment, but the mystery attaching to its effects should be a sufficient excuse for most carefully investigating all such cases as may possibly have been produced by it.

Influence of Punching.—One treatment which produces injurious effects in steel is punching holes into the plates. The experiments on the subject are too numerous to be mentioned here, but many of them will be found in the chapter on 'Mechanics,' under 'Riveted Joints,' p. 169. The thickness of the plate, the diameters of the punch and die, as well as the hardness and chemical composition of the plate, affect the pliability of such samples. It has been found that rimering out $\frac{1}{16}$ in. of the holes removes all bad effects; but if it is true, as stated by Mr. Beck-Gerhard ('Gorni J.,' 1884, p. 347), that the curves of stress slowly extend at least 5 ins. away from the hole, then punched plates ought to be rimered at once, or annealed. This experiment was as follows:—A $\frac{3}{8}$-in. plate was polished on one side and punched (in a cold atmosphere). Spiral curves then showed themselves, which were first washed with aqua regia. The piece was then planed into several strips and each tested, when the spiral curves reappeared and were perceptible to the touch.

This may explain the curious curved markings near punched holes and sheared edges. In every one of these lines the surface scale has fallen off, showing that here stresses have been at work producing local deformations of at least $\frac{1}{2}$%, for it is only after steel has been stretched this amount that the mill scale falls off the plates. Illustrations of similar effects will be found in Kirkaldy's works. (See p. 127.)

Influence of Severe Stresses.—An important matter is the behaviour of steel and iron under severe stresses, and a good deal may be learnt on this subject by watching a tensile test piece under the following conditions: After being fixed in the machine a strain indicator is attached, and the loading carefully proceeded with, preferably by the addition of small loads, and not by moving a jockey weight. The elongation produced can then be accurately reduced to zero by removing the weights.

Viscosity.—It will be found that even with small loads a very small permanent set takes place. This is generally attributed to viscosity of the material, and is best studied on wires subjected to torsion. Pitch possesses this quality in a high degree, and quartz is said to be absolutely free from it; probably it is this property which makes a metal non-sonorous, from which it would follow that hard steel, silver, and glass are not of a viscous nature. H. Tomlinson, 'Phys. S.,' 1887, vol. viii. p. 171; 1888, vol. ix. pp. 49, 67.

Modulus of Elasticity.—The loading of the test piece can now proceed, careful readings of the strain indicator being taken. It will be noticed that for steel and iron the elongation is almost proportional to the stress, being at the rate of $\frac{1}{1000}$ in. in 10 ins. for about every 1·3 ton per square in., from which it follows that the modulus of elasticity is about 13,000 tons, or 30,000,000 lbs. per square in., or 20,000 kils. per square millimeter. With cast iron and various other metals

there seems to be a change in this modulus when the stress is increased beyond a certain point, and some people have called this the limit of elasticity, but a better name is limit of proportional elongation. The more accurate the strain indicators are, the more gradual does this change appear, and the obvious conclusion is that with these metals the modulus of elasticity is a variable quantity, growing smaller as the stress increases. It also decreases about $1\frac{1}{2}$ per cent. for every 100° F. rise of temperature (see H. Tomlinson, 'Phys. S.,' 1887, vol. viii. p. 171).

Elastic Limit.—It will be noticed that the pointers of the strain indicators oscillate slightly for every newly-added load, coming to absolute rest only after a very long period. But when a certain stress has been reached this action ceases and the pointers acquire a slow onward motion. The elastic limit has now been reached, and to verify whether this is so or not several or all the weights are removed, and the pointers will either return to the positions previously occupied or not. This check is necessary, as the giving way of the attachments of the test pieces sometimes produces strange effects, and may even cause one of the pointers to travel backwards. For this reason also the determination of the elastic limit should always be made with the help of three strain indicators. Mild steel shows an elastic limit of about 15 tons. If very mild and previously tempered it is sometimes as low as 10 tons, but in such cases it is hardly perceptible and no breakdown point can be noticed.

Breakdown Point.—Continuing the straining of properly annealed samples, a point will be reached when the slow motion of the pointers gives way to a very rapid one, which is not stopped even by reducing the stress by several tons. This is called the 'breakdown point,' because of the behaviour of the material, or the 'drop,' the lever falling through a considerable angle. It has also been called the limit of plasticity, because above this point the material behaves as if it were plastic. This point is often mistaken for the limit of elasticity, and the two points sometimes fall together.

Irregular Stretching.—Even now it is of interest to watch the strain indicator, for it will be found that the plastic elongation proceeds very irregularly. Generally after adding a weight it commences slowly, increases, and then diminishes, until at last no further motion can be detected. Very often, particularly if only small additions are made to the load at one time, the pointers vary their speed repeatedly, increasing and diminishing their velocity several times without any additional weights being added to the lever.

Irregular Elongations.—An explanation will suggest itself if several (say, four) short strain indicators are attached along the length of the sample, for it will then be noticed that first one and then the other span is elongating, showing that waves of plasticity pass along the samples. This may also be noticed when the breakdown point is reached, for then the mill scale falls off, first at the extremities, and then more towards the centre. The scale falls off when the stretch exceeds $\frac{1}{2}$ % of the length.

The same phenomena, but more marked, may be noticed when twisting wire in a torsion machine. Instead of proceeding uniformly, it will be noticed that the twist commences at one end (fig. 115), and that, like a wave, it travels to and fro till the sample breaks.

Changes in the Limit of Elasticity.—The next point to be noticed in a test piece is that when once the elastic limit has been passed, and the sample then unloaded, it will not again elongate permanently until the previous stress has been reached. This is only natural, for the

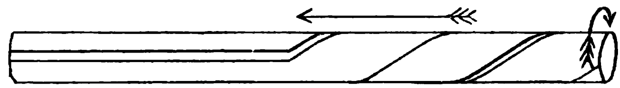


FIG. 115.

second testing is but a continuation of the first, and it is not difficult to accept the statement that the elastic limit of a sample is raised by preliminary testing.

R. H. Thurston, 1883, p. 601, has suggested that this behaviour would enable one to detect whether a broken structure had been overstrained. A test piece would have to be carefully cut from the plate and its limit of elasticity determined. Unfortunately experiments show that, even with the most careful handling during preparation, the elastic limit of the test piece has again fallen to its original value, or perhaps the shock of the rupture has produced this effect.

Influence of Time.—The test piece under consideration could now be stretched till it has elongated 5 %, which will raise the stress to about 25 tons. It may then be left in the machine overnight with the full load on it, or it may be put aside for a day or two. On re-testing it will be found that the elastic limit has risen considerably above 25 tons, which was the last stress to which it was subjected.

The following experimental results will illustrate this :—

Sample No. 1	
Elastic limit	19·3 tons
Ultimate strength	29·7 tons, 20·1 % elongation

Sample No. 2	
First elastic limit	16·3 tons
Then loaded to	24·6 tons, 5·2 % elongation
Second elastic limit . . .	28·5 tons after an interval of 10 days
Ultimate strength	29·8 tons, 17 % elongation

A long list of experiments on this subject by Bauschinger is contained in 'Civil I.,' 1881, vol. xxvii. p. 1 ; also 'Mitt. Munich,' 1886, vol. xiii. He investigated the behaviour of 14 samples of iron and steel, and also copper and gunmetal. Unlike the author's experiments, none of his showed an increase of elastic limit beyond the ultimate strength of the material, but even with him the influence of time in raising it is very marked. With copper and gunmetal the elastic limits only rise as high as the preliminary stress. In all these cases the second elastic limit and breakdown point fall together, and the drop is now very much greater than with an annealed sample.

The question naturally arises, What would happen if the sample had first been subjected to a compression test? This experiment has also been carried out, and it was found that the elastic limit for ten-

sion had been reduced from 19·3 and 16·3 to 11·8 tons, and the ultimate tenacity raised to 30·7 tons, elongation 7·2 %.

A preliminary compression stress at right angles to the axis of the sample (produced by drawing it out under a hammer) raised the limit to 20·5 tons. This also increased the tenacity to 32 tons, and reduced the elongation to 12 %.

These experiments readily suggest that, as the elastic limit is a changeable value, it cannot be a reliable measure of the working strength of a material. When a preliminary proof test has raised the elastic limit considerably it may be dangerous to repeat it, because if the new elastic limit is accidentally exceeded a very considerable breakdown occurs, which may lead to rupture. Those parts of a structure which have been subjected to excessive compression stresses should not be exposed to severe tension stresses, as their elastic limits of tension have been lowered and their ductility reduced. The reverse is also probably true.

Fatigue.—Closely related to the last proposition are the deductions drawn from Wöhler's celebrated experiments on the effect of alternating stresses, better known by the name of fatigue.

They have been and are still being repeated, and it would appear—

Firstly, if the experimental tension and compression stresses are sufficiently low, these may be repeated an infinite number of times without producing rupture.

Secondly, rupture will be produced by alternate stresses, if these are sufficiently high, but kept below the elastic limit.

Thirdly, the more intense the alternate stresses are, the sooner will rupture occur.

Fourthly, if the alternate stresses are equal in intensity, they produce rupture more quickly than if one of them is small or does not exceed zero.

Attempts have been made to find a relation between the power of a metal to resist fatigue and its other known qualities, such as elastic limit, strength, and elongations, but as yet these are not reliable.

A good deal of information on this subject will be found in the following publications:—Wöhler, 'Zeit. Bw.,' 1860, vol. x. col. 583; 1863, vol. xiii. col. 243; 1866, vol. xvi. col. 67; 1870, vol. xx. col. 90. Spangenberg, 'Zeit. Bw.,' 1874, vol. xxiv. col. 482; 1875, vol. xxv. col. 79. Ibid. 'Glaser's An.,' 1879, vol. v. col. 6.

The various theories based on these and other experiments are to be found in the following papers:—Dr. J. Weyrauch, 'C. E.,' 1880, vol. lxiii. p. 283; Gerber, 'Bay. A. I. V.,' 1874; Lippold, 'Organ,' 1879, vol. xvi. p. 22; Launhardt, 'Arch. I. V.,' 1873; Professor Mohr, 'Civil I.,' 1881; H. Tresca, T. Seyrig, E. E. Marchée, E. Trélat, A. Brull, H. Mattieu; 'Ing. Civ.,' Résumé, 1881, vol. ii. p. 39.

Contraction of Test Pieces.—Continuing to follow the behaviour of a test piece, and this time until fracture takes place, it will be found that some materials, such as hard steel, manganese steel, bad iron, and gunmetal, show no contraction previous to rupture, while mild steel, iron, and brass and copper, do. The reasons for this contraction are not known. It has been suggested that contraction is due to local weakness, so that those metals which contract most are least uniform as regards tenacity; but if that were the case tensile tests with drilled

samples would be very irregular, according as to whether the hole was near a weak or a strong place. The following experiments on a mild steel plate whose tenacity was 28 tons show that this is not the case :—

The samples were shaped as shown in figs. 116, 117. Thickness of plates $\frac{1}{2}$ in.; diameter of hole 1 in., drilled.

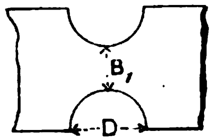


FIG. 116.

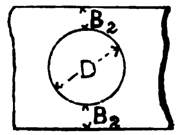


FIG. 117.

Breadth		Ult. Strength
B_1. Inches	B_2. Inches	Tons per Square Inch
1·85	...	30·1
1·25	...	30·5
·71	...	30·3
...	·53	29·6
·49	...	30·5
...	·43	29·8
...	·33	29·1
...	·28	28·6
...	·23	30·5
...	·18	28·7
...	·13	28·9
...	·075	28·6

(See also E. Richards, 'I. and S. I.,' 1882, p. 43.)

Another suggestion is, that the rise of temperature of a test piece weakens the part which first contracts more than the others which have accidentally not contracted. Dr. J. Kollmann's experiments ('Ver. Gew.,' 1880, 2nd ser. vol. lix. p. 104) confirm this, for there it will be found that the contraction steadily increases from about 20 % at an ordinary temperature to 90 % at a red heat. Unfortunately for this view the tenacity does not show the same regularity. An explanation of this phenomenon is, therefore, still required. The subject will be referred to again when discussing compound stresses.

Fractures.—The next thing to be noticed in a fractured test piece of mild steel is, that when placed together only the edges touch, leaving a hollow, as shown in fig. 118. There can be but one explanation, viz. that the rupture started near the centre of the section, and that the outside fibres continued to stretch after this point was reached. This suggests the view that the stresses to be found in the centre of a test piece differ from those on the outside surfaces, and are also more injurious. An examination of almost any torn sample of mild steel will show that near the edge the surfaces of rupture are very much on the slant and have every appearance of having been produced by partial shear, suggesting the idea that this material gives way more readily under shearing stress than under tension (see p. 126).

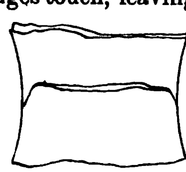


FIG. 118.

The same tendency is noticed when tearing a sample perforated as in fig. 119. The lines of fracture, instead of running as horizontal as possible, will be distinctly steeper than the angle at which the holes were drilled.

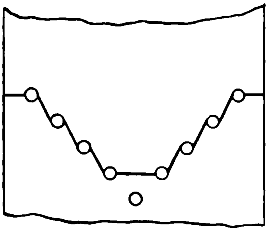


FIG. 119.

A few other matters which have been noticed in fractures are: White specks, which are due to a local excess of phosphorus and other impurities (Stubbs, 'I. and S. I.,' 1881, p. 379; H. Eccles, ibid., 1888, p. 72; Prof. Ledebur, 'Stahl und Eisen,' 1889, vol. ix. p. 13). A smell of ammonia, said to be due to occluded nitrogen, is sometimes noticed; the colour also varies from bluish grey to salmon-colour tints or yellow ones.

Shearing Stresses.—The investigation of shearing stresses is beset with various difficulties, one of the most important being the smallness of the strains. Direct experiments have therefore only been useful in determining the ultimate shearing strength of various materials.

It has been found that the hardness of the metal into which the holes are drilled influences the results. It is therefore usual to groove the samples to be tested as shown in fig. 120. Most experiments on riveted joints include some carried out as above, and the general conclusion is that the shearing strength is less than the tensile strength.

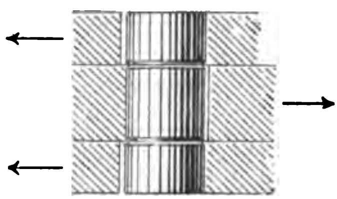


FIG. 120.

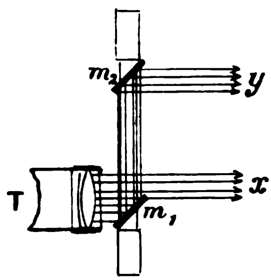


FIG. 121.

More interesting results are obtained if the material is exposed to torsion stresses. One end of a cylindrical bar is secured to the head of a lathe, while the other end is supported, if possible, on a knife-edge, and carries a lever, which is gradually loaded while the lathe head is being turned round. The twisting of the bar is measured with the help of a pair of mirrors m_1, m_2, arranged as shown in fig. 121. A graduated scale is placed at a considerable distance from the test piece, and examined partly direct (x), and partly by doubly reflected light (y). Two scales instead of one are then visible through the telescope T, and it is their relative displacement which measures the angular deflection. This arrangement has been devised by the author and has proved to be very accurate.

By plotting down the readings, curves are obtained, which may be called torsion diagrams, and represent the amount of twist which

various torsion moments impart to a test bar. It is usual to reduce all these values so as to obtain the shearing stresses of the outside fibres and their angular displacement.

The latter value is found by multiplying the angle of twist into half the diameter of the bar, and dividing by the distance of centres.

The shearing stress of the outside fibre is found by the formula

$$\sigma_0 = \frac{16 \cdot M}{\pi \cdot d^3}.$$

Here M is the torsion moment, and d is the diameter of the bar. This formula is only correct as long as the moments are strictly proportional to the twist; where this is not the case, and particularly when the limit of elasticity is passed, or the point of rupture reached, it gives wrong results.

The actual shearing stress σ in the outer fibre is then found as follows: Let the heights of the line O B C (fig. 122) represent the distribution of the circumferential shearing stresses over the radius of a bar whose diameter is $d=2r$, and which has been twisted through an angle θ in a length l, while the torsion moment is M.

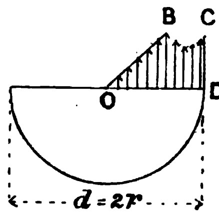


FIG. 122.

D C represents the shearing stress in the outer fibre. Now if a thin film of metal, of the thickness dr, be machined off the circumference of the bar, and if it be twisted once more, till the stress in the (now reduced) outer fibre is again equal to D C, then, as the new curve O B C is similar to, but shorter than, the original one, the new twisting moment M_1 will have to be somewhat weaker than M, viz. $M_1 = M \cdot \frac{(r-dr)^3}{r^3}$, while the angle θ has increased from θ to $\theta + d\theta = \theta \cdot \frac{r}{r-dr}$. Now let a film of material, of the thickness dr, be placed round the bar, so that it is once more equal to its original diameter, $2r$. Let this outer film be twisted till it has acquired a stress of $\sigma + \frac{d\sigma}{dr} \cdot \frac{dr}{2}$; then the torsion moment of the film is $\left(\sigma + \frac{d\sigma}{2}\right) 2 \pi r^2 dr$, from which it follows that

$$\frac{dM}{dr} = -3 \cdot \frac{M}{r} + 2 \cdot \pi \cdot r^2 \cdot \sigma.$$

But, as $\frac{dr}{r} = \frac{d\theta}{\theta}$, we have

$$\sigma = \frac{1}{2 . \pi . r^3} \cdot \left(3 \cdot M + \theta \frac{dM}{d\theta}\right) = \frac{4}{\pi . d^3} \left(3 \cdot M + \theta \frac{dM}{d\theta}\right),$$

which formula can easily be converted into $\sigma = \frac{3}{4}\sigma_0 + \frac{\theta}{4}\left(\frac{d\sigma_0}{d\theta}\right)$.

The way to construct a curve which will show the actual shearing stress σ of the outside fibre, when the value $\sigma_0 = \frac{16 . M}{\pi . d^3}$ has been previously determined, can be carried out as shown in fig. 123. The line σ_0 is copied from Platt and Hargraves's paper ('C. E.' 1887, vol. xc. plate 10, fig. 6). The faint tangential lines have been drawn to measure the

value $\frac{d\tau_0}{d\vartheta}$. The line $\frac{3}{4}\sigma_0$ has been constructed by proportional reduction, and to this has been added the respective values of $\frac{\theta}{4} \cdot \frac{d\sigma_0}{d\vartheta}$, with the help of which the curve σ has been constructed. The experimenters state that for this particular sample the elastic limit of shearing stress

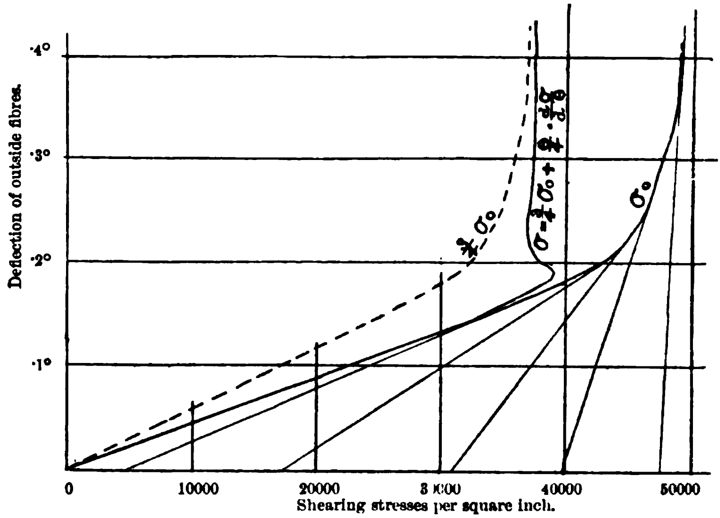


FIG. 123.

was 46,400 lbs.; but an examination of the curve σ shows that it was already reached at 35,000 lbs., and that at 38,000 lbs. a very serious drop took place, and the material of the outside fibre had not recovered even when the deflection had increased to $\frac{1}{2}°$. Beyond this point it is always sufficiently accurate to adopt the value $\sigma = \frac{3}{4}\sigma_0 = \frac{12 \cdot M}{\pi . d^3}$. That this is fairly correct will be seen from the following experiments on tension, torsion, and direct shear (Platt and Hargraves, 'C. E.,' 1887, vol. xc. p. 408):—

Materials	Limits of Elasticity		Ultimate Strengths		
	Tension	Estimated Shear $\sigma = \frac{16 \cdot M}{\pi d^3}$	Tension	Estimated Shear $\sigma = \frac{12 \cdot M}{\pi d^3}$	Experimental Shear on Rivets
	Tons	Tons	Tons	Tons	Tons
Wrought iron .	15·04	8·99	21·60	17·63	18·76
" " .	16·82	10·36	25·00	20·20	21·21
" " .	17·13	10·22	24·56	20·65	20·72
Bessemer steel .	31·14	20·28	52·20	31·25	35·21
Crucible " .	31·06	19·36	52·16	29·65	33·30
Cast " . .	17·22	10·40	38·04	24·30	27·60
Siemens " .	17·85	10·20	28·40	20·90	23·00
" " .	16·82	10·16	25·75	19·70	21·05
Muntz metal . .	11·20	8·70	25·46	18·26	18·60
Gunmetal Cu 64, Sn 8, Zn 2	7·25	5·40	13·68	11·06	12·47

These, D. Kirkaldy's, and some similar experiments by G. Berkley, 1868, V. Appleby ('C. E.,' 1883, vol. lxxiv. p. 268), the latter including compression tests, are apparently the most exhaustive ones that have yet been made to ascertain the relation which exists between compression, tension, and shearing stresses.

The latter are certainly always smaller than either of the two former, amounting to from 50 to 90 %.

Torsion tests carried out on bars from which the black scale had not been removed showed that it falls off when the limit of elasticity is reached, but even then only along a few axial lines, which implies that the elastic limit and breakdown point fall together.

Bending Stress.—After the foregoing it will be unnecessary to analyse the behaviour of beams subjected to stresses beyond their elastic limit; suffice it to say that as the limit of elasticity of the top and bottom fibres are not necessarily reached at the same time, it is impossible to separate the one from the other; but, assuming that they did agree, then the stress in the outside fibre of a narrow, rectangular bar would be found by the following formula:

$$S = \frac{6}{h^2 b}\left(\tfrac{2}{3}M + \frac{c}{3}\frac{dM}{dc}\right).$$

Here c stands for curvature.

From an examination of fig. 124, which represents a beam subjected to an excessive load, Q, it is evident that $\frac{dM}{dx} = Q$, and, by carefully measuring the various curvatures and changes of curvatures of such a sample, it would be possible to construct a strain stress diagram with the help of the following formula:

$$S = \frac{Q}{h^2 . b} \cdot \left(4 . x - 2 . \rho . \frac{dx}{d\rho}\right).$$

It must not be forgotten that ρ, the radius of curvature, grows smaller with increasing x, and that $\frac{d\rho}{dx}$ is negative.

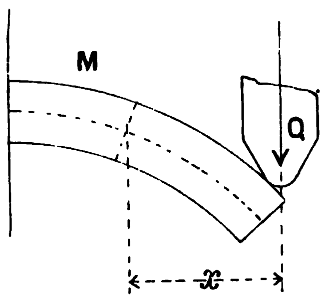


FIG. 124.

As in the case of torsion experiments, the second term in the bracket is very small when the point of rupture is reached, and to estimate the ultimate strength of a plastic material, including cast iron, but not glass or other brittle substances, it is sufficiently accurate to use the formula

$$S = \frac{4.Q.l}{h^2 . b}.$$

This is 33 % less than obtained by the generally accepted elastic formula, and accounts for the fact that beams are apparently so much stronger than they should be.

Compound Stresses.—It has already been pointed out that the elastic limit and ultimate strength of a material are very much less in shear than in tension, or in compression. But it is well known that a **shearing stress** is composed of a tension and a compression stress,

each of equal intensity, as shown in fig. 125, and it is therefore of importance to ascertain what other compound stresses exist.

There are, firstly, the simple stresses, (I.) **tension**, and (II.) **compression.**

A combination of two of these acting at right angles produces a (III.) **shearing stress** (fig. 125).

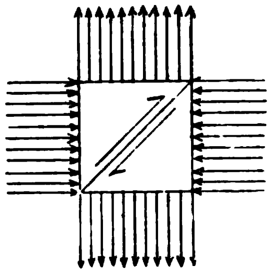


FIG. 125.

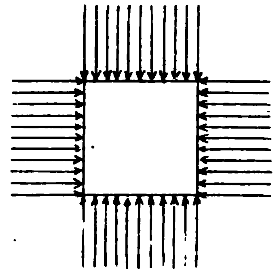


FIG. 126.

IV. Two equal compression stresses are met with chiefly in railway axles when the wheel boss has been shrunk on them. This combination might be called a **shrinking** or **strangling stress**, to distinguish it from a compression stress, which acts only in one direction. (See fig. 126.)

V. The reverse of this stress might be called **drum tension** (fig. 127), as it is best represented by that case; it is also met with in thin spherical shells subjected to internal pressure.

By adding stresses at right angles to the planes in which the last two are acting four others are obtained.

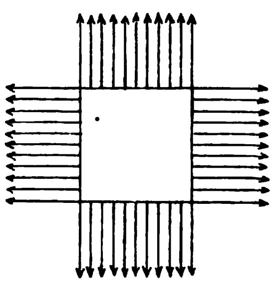


FIG. 127.

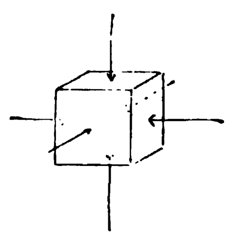


FIG. 128.

VI. **Fluid Pressure** (fig. 128).—In this case there are only compression stresses. If they are all changed into tension stresses we get a combination which might be called (VII.) **negative fluid pressure**, or **solid tension** (fig. 129).

VIII. When two of the stresses are compression, and the other one tension, we have the case of wire-drawing. This might be called a **draw stress** (fig. 130). (See p. 111.)

IX. By combining two tension stresses with one compression we reproduce a condition which is found on the inner spherical surfaces of very thick-walled exploding shells. This might be called a **bomb stress**

(figs. 131 and 181). Of the last six combinations there is only one about which anything definite is known, and that is that no material, however weak it may otherwise be, has been destroyed by fluid pressure, however great.

Solid tension (or negative fluid pressure) does not occur in practice, but the following two cases are an approach to this condition.

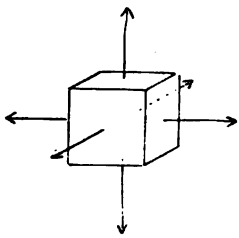


FIG. 129.

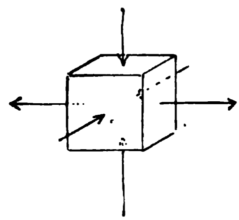


FIG. 130.

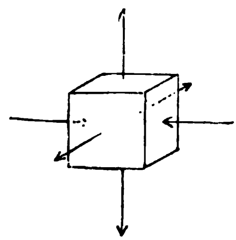


FIG. 131.

If a solid sphere is heated to redness, and then plunged into cold water, the outer surface solidifies while the centre is still red hot. The external diameter will be somewhat larger than it would have been if cooled slowly; and when the centre has grown cold, a tension will be found there acting in every direction. In large masses of steel this tension even comes into existence while the centre is still red hot, and cavities are formed, to prevent which ingots are never cast circular, but square or of polygonal shape, so that the sides may collapse. Of course it is impossible to estimate the stress which has produced these holes.

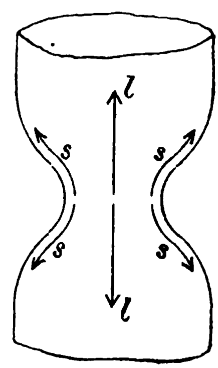


FIG. 132.

A somewhat similar stress is found at the point of contraction of test pieces (fig. 132) when of a circular section. The sample is being stretched in the direction of the arrows, l, l, and the lines of force, s, s, will adapt themselves to the fibres, at any rate at the circumference, and there their curved shape will produce radial tensions, as indicated by the looped arrows (fig. 133), so that at the centre of the smallest section there exists a tension in every direction.

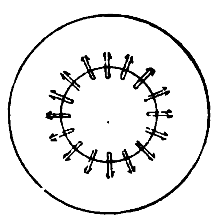


FIG. 133.

This might explain why with mild steel, where there is considerable contraction of area, the fracture starts at the centre, the material being less able to withstand a solid tension than a simple one, as at the circumference of the fracture. A careful analysis of the distribution of stresses produced by the load at the instant of rupture will perhaps enable one to obtain numerical values for different materials. Thus, it is not unusual for the contraction to exceed 50 %, and if the load at rupture was 80 % of the maximum, or, say, 24 tons per square in. instead of 30, the average stress in the reduced section must have been 48 tons; and when this is combined with the drum tension, due to the

shape, which also exists there, it is not unreasonable to assume that the sample only gave way to a solid stress whose components amounted to from 96 to 144 tons. Hard cast steel will not resist these compound stresses without rupture, and therefore does not contract; and it might even be questioned whether for compound stresses it is as strong as the milder qualities.

Of the other compound stresses little is known, except, perhaps, that a draw stress has a very marked effect on lowering the elastic limit of materials, for at present no other explanation will account for the possibility of drawing wires.

Until experiments have been made to determine the permissible compound stresses it would be useless to speculate as to their action, but enough has been said to show that we are as yet groping in the dark. It is even impossible to say whether a spherical boiler end may be made half as thick as the cylindrical shell; for although theory shows that the stress is only one-half, it also shows that there are two equal stresses, which have been called drum tension, and, judging by what takes place under a shearing stress, it is not unreasonable to assume that a drum tension is twice as injurious as a simple tension.

CHAPTER VI.

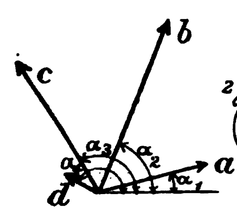
MECHANICS.

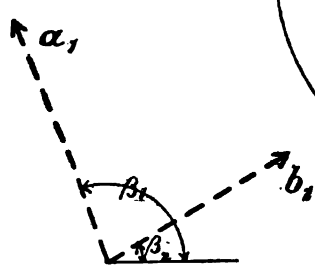
INVESTIGATIONS in statics show that several forces acting through one point can always be replaced by a single resultant, and similarly several stresses can be replaced by resultants; but there will be three instead of one, unless they are all acting parallel to one plane, and in that case there will only be two. To resolve stresses which are irregularly distributed in space is a problem which need not be discussed in this chapter, particularly as it leads to rather complicated formulæ (Rankine, 'R. Soc. Edinburgh,' vol. xxvi. p. 715).

The Resolution of Stresses parallel to one plane is comparatively simple.

Let there be four stresses, a, b, c, d (fig. 134), acting in the directions α_1, α_2, α_3, α_4. Double each of these angles and construct the polygon A, a, b, c, d, B (fig. 135). The points A, B may then be looked upon as the

FIG. 134.





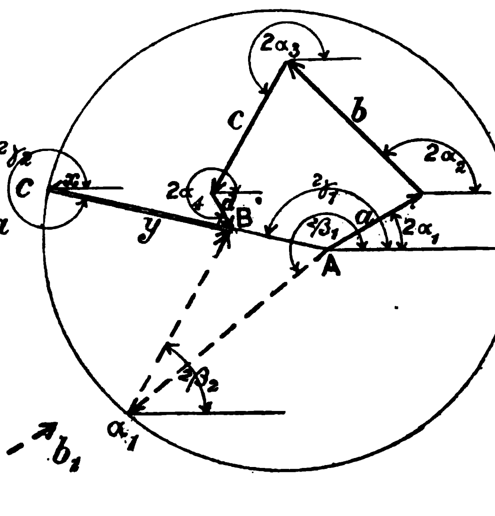


FIG. 136.

FIG. 135.

two foci of an ellipse, the sum of the radii vectores A, α_1, B being equal to the sum of the stresses. By halving the angles $2\beta_1$ and $2\beta_2$, and drawing the stresses a_1 and b_1 from a point (fig. 136), a system of two resultants

is obtained which would produce exactly the same strain as the four original stresses; but they are not necessarily at right angles to each other. In order to fulfil this condition, radii vectores ACB (fig. 135) will have to be drawn parallel to the major axis, and as $2\gamma_2 = 2\gamma_1 + 180°$ the resultants will be at right angles to each other, as shown in fig. 137. Their intensities are represented by the lengths AC$=x$ and CB$=y$.

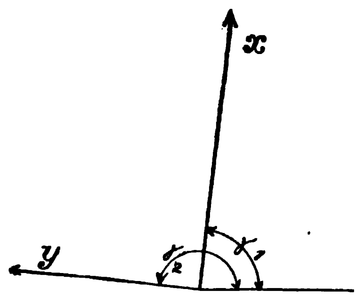


FIG. 137.

The following is the algebraical solution of this question:—

The notations are the same as in the figures.

As regards the angles we have—

$$\text{AB} \,.\, \sin 2\gamma_1 = a \sin 2a_1 + b \sin 2a_2 + \&c.$$
$$\text{AB} \,.\, \cos 2\gamma_1 = a \cos 2a_1 + b \cos 2a_2 + \&c.$$
$$\tan 2\gamma_1 = \frac{a \sin 2a_1 + b \sin 2a_2 + \&c.}{a \cos 2a_1 + b \cos 2a_2 + \&c.}$$

Divide $2\gamma_1$ by 2; this determines γ_1; and add 90°, which is the angle γ_2.

As regards the stresses we have—

$$x = \frac{(a+b+\&c.)}{2} + \frac{\text{AB}}{2} = \frac{\Sigma(\text{S}) + \sqrt{\Sigma(\text{S} \,.\, \sin 2a)^2 + \Sigma(\text{S} \,.\, \cos 2a)^2}}{2}$$
$$y = \frac{(a+b+\&c.)}{2} - \frac{\text{AB}}{2} = \frac{\Sigma(\text{S}) - \sqrt{\Sigma(\text{S} \,.\, \sin 2a)^2 + \Sigma(\text{S} \,.\, \cos 2a)^2}}{2}.$$

If some of the stresses are positive and the others negative, i.e. tension or compression, they should be placed in proper order in the polygon (fig. 135). But then, instead of an ellipse, an hyperbola will be the boundary line. It can also be proved that there will only be one resultant to several shearing stresses. Thus, if in fig. 134 they are represented by a, b, c, d, the intensity of the resultant stress would be AB (fig. 135) and its angle γ_1.

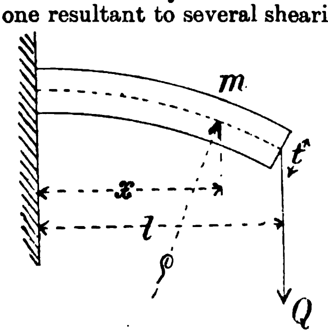


FIG. 138.

The Elastic Beam.—When loaded at one end (fig. 138) the bending moment of a beam, at the distance x from its attachment, is

$$m = \text{Q} \,.\, (l - x).$$

If the longitudinal stresses at this point vary from $-$ S at the bottom fibres to $+$ S at the top fibres, they combine to form a resisting moment which is equal to m, which for a rectangular section is:

$$m = \frac{\text{S} \,.\, t^2 b}{6}.$$

Here b is the breadth of the beam, and t its thickness. For a round bar of the diameter d we have

$$m = \frac{\pi \cdot S \cdot d^3}{32}, \text{ or nearly } \frac{S \cdot d^3}{10}.$$

In flat plates b can be taken as unity, provided the pressures and forces are measured per unit of width.

The Radius of Curvature ρ can be determined from the following equation:

$$\frac{1}{\rho} = \frac{d^2y}{dx^2} = \frac{m}{E \cdot I} = \frac{2 \cdot S}{t \cdot E}.$$

Here x and y are the co-ordinates of any point of the elastic line, E is the modulus of elasticity, and I is the moment of inertia of the section $\left(I = \frac{t^3}{12} \text{ for a flat beam 1 in. wide}\right)$.

Cross Curvature.—Besides being bent lengthways, the edges of the beam curl up (fig. 139). If the radius of this curvature is called ρ_1 it will be found that

$$\frac{\rho}{\rho_1} = \frac{1}{\mu} = \text{coefficients of cross contraction.}$$

There are various means for determining it, one of the simplest being to compare the modulus of shearing elasticity E_1 with that due to direct tension.

$$E = E_1 \cdot 2\left(1 + \frac{1}{\mu}\right).$$

Unfortunately there is little agreement amongst the results of experiments, carried out, as they usually are, on wires; but as the most reliable ones show $\frac{1}{\mu}$ to be about $\frac{1}{3}$, that value will be adopted here wherever numerical results are given.

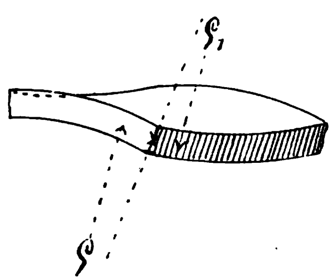


FIG. 139.

The cross curvature should not be neglected when accurate measurements are being taken for the purpose of ascertaining the stresses existing in a beam; for if it is a very wide one, or other means are adopted to prevent its edges from curling up, it is quite clear that cross stresses are being set up, which in their turn affect the longitudinal deformation.

Suppose that $\rho_1 = \infty$, then

$$\frac{1}{\rho} = \frac{2}{t \cdot E}\left(S - \frac{C}{\mu}\right) = \frac{2 \cdot S}{t \cdot E}\left(1 - \frac{1}{\mu^2}\right).$$

Here C is the maximum cross stress set up in the beam

$$C = \frac{S}{\mu}.$$

If the elastic curvature of a flat end plate of a boiler between the stays,

or of an unstayed flat dome end, is found to be equal in every direction, and if its radius is ρ, then the following relation exists :

$$\frac{1}{\rho} = \frac{2 \cdot S}{t \cdot E}\left(1 - \frac{1}{\mu}\right).$$

So that if this radius had been used for the determination at the stress S by the first formula, a mistake of 33 % would have been made.

The correct formula is

$$S = \frac{E \cdot t}{2 \cdot \rho} \cdot \frac{\mu}{\mu - 1}, \text{ or about } \frac{E \cdot t}{2 \cdot \rho} \cdot \frac{3}{2}.$$

At those points where the cross stresses are equal a drum tension exists on the convex side of the plate, and a shrinking stress on the concave side (see p. 128) ; but, as the former will most likely injure the material more seriously than a simple tension (see p. 130), it is very important that all measurements of curvature made for the determination of stresses should be taken both lengthways and crossways. Conversely, a case might occur where $\rho_1 = - \rho_2$; then $S = \frac{E \cdot t}{2 \cdot \rho} \cdot \frac{\mu}{\mu + 1}$, showing that the stresses are smaller than usually supposed. Combined in this way they constitute a shearing stress of a very peculiar nature, being directed to opposite directions on either side of the plate. This condition is found to exist along the pitch lines of screwed stays, near the centres of the pitches. This cross curvature of the beams naturally affects their deflections, and in it is to be sought the explanation why it never agrees with what (so-called) theory makes it out to be.

Stresses in Flat Plates.—The formulæ which would have to be evolved in analysing these stresses are very complicated, even when dealing only with circular discs, and as such results are of very limited value for these investigations, and as analyses of square and rectangular plates leave out of account some important points, a short explanation as to the methods adopted is all that can be attempted here. Fig. 140 is a section through the centre of a circular disc of the diameter $2r$ and of the thickness t. It is loaded with a pressure p over its entire surface, and with a load Q spread over its circumference, and is supported by a central stay. Certain strains and stresses will be produced, which it is desired to determine.

Imagine the disc to be subdivided by a large number of radial lines ; then one of the slices so formed could be represented by fig. 141. Its angle is β. It is probable that radial bending moments exist, and if the maximum radial tension stress of the upper fibres at the distance x is denoted by S the resisting moment will be

$$m_1 = \frac{S \cdot x \cdot \beta \cdot t^2}{6}.$$

If now the disc is subdivided into a number of concentric cylinders, the thickness of whose walls is dx (figs. 142, 144), it will be found that under stress they have lost their original cylindrical shape, and have grown slightly conical. Let this angle be α. It is clear that circumferential stresses exist in these rings, tension at the upper surface, and

compression at the lower ones. Let the maximum value of the former be C. These stresses could also have been produced by an imaginary internal fluid pressure, ranging from $+dp$ (fig. 143) at the upper surface, to $-dp$ at the lower.

If now a short length, $\beta . x$, of the ring be examined, the tilting power of this imaginary fluid pressure would be found to be

$$dm_2 = dp \,.\, x \,.\, \beta \,.\, \frac{t^2}{6};$$

but, as $-\mathrm{C} . dx = +dp . x$, we have

$$-\frac{dm_2}{dx} = \frac{\mathrm{C} \,.\, \beta \,.\, t^2}{6}.$$

The intensities of S and of C can, therefore, be expressed in terms of m_1 and m_2. The relation existing between these moments is determined by the deformations of the disc.

The differential equation for the elastic line is $\frac{d^2y}{dx^2} = \frac{m_1}{\mathrm{E} . \mathrm{I}}$, as long as there are no cross stresses; but, as the circumferential tension C tends to shorten the radial fibres, the coefficient of cross contraction $\frac{1}{\mu}$ must be introduced, and the equation of the elastic line is

$$\frac{d^2y}{dx^2} = \frac{2}{t . \mathrm{E}}\left(\mathrm{S} - \frac{\mathrm{C}}{\mu}\right).$$

α, the angle of the cone, is equal to $\frac{dy}{dx}$, which, by a similar reasoning to that of the previous case, is

$$\frac{dy}{dx} = \frac{2 . x}{t . \mathrm{E}}\left(\mathrm{C} - \frac{\mathrm{S}}{\mu}\right).$$

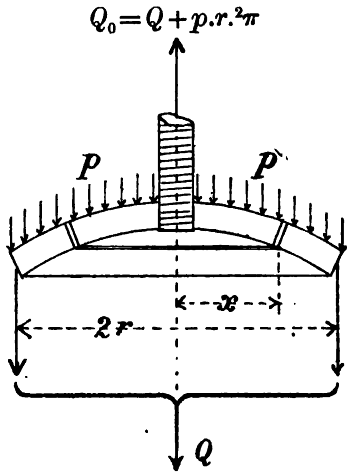


FIG. 140.

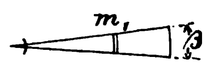


FIG. 141.

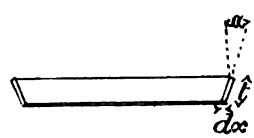


FIG. 142.

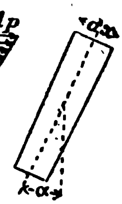


FIG. 143. FIG. 144.

Differentiating this, we get

$$\frac{d^2y}{dx^2} = \frac{2}{t . \mathrm{E}}\left(\mathrm{C} - \frac{\mathrm{S}}{\mu} + x . \frac{d\mathrm{C}}{dx} - \frac{x}{\mu} . \frac{d\mathrm{S}}{dx}\right).$$

Combining this with the previous value for $\frac{d^2y}{dx^2}$, it disappears, and

we get

$$O=(C-S)\left(1+\frac{1}{\mu}\right)+x\left(\frac{dC}{dx}-\frac{1}{\mu}\cdot\frac{dS}{dx}\right).$$

This is the fundamental equation as regards the relation of the two stresses C and S. These have now to be compared with the external forces producing them.

For simplicity's sake let the perforated plate (fig. 140) be suspended by a stay of the diameter d, and uniformly loaded only at its circumference by Q. Dealing once more with a narrow slice of the angle β, it will be found that at the point x there exists a vertical shearing force $q=\frac{Q\cdot\beta}{2\cdot\pi}$, which acting on the lever $x-\frac{d}{2}$ produces a moment

$m_0=\frac{Q\cdot\beta}{2\cdot\pi}\left(x-\frac{d}{2}\right)$, and therefore $\frac{dm_0}{dx}=\frac{Q\cdot\beta}{2\cdot\pi}$.

The moment due to the radial stresses is

$$m_1=\frac{\beta\cdot t^2}{6}\cdot x\cdot S,\ \text{and}\quad \frac{dm_1}{dx}=\frac{\beta\cdot t^2}{6}\left(S+x\frac{dS}{dx}\right).$$

The elementary moment, dm_2, produced by the circumferential stresses C, has to balance dm_0+dm_1.

$$-dm_2=\frac{\beta\cdot t^2}{6}\cdot C\cdot dx\ .=\left\{\frac{\beta.t^2}{6}\left(S+x\frac{dS}{dx}\right)+\frac{Q.\beta}{2.\pi}\right\}dx,$$

from which it follows that

$$C=S+x\cdot\frac{dS}{dx}+\frac{Q\cdot 3}{\pi\cdot t^2}.$$

By differentiating and substituting the values of C and $\frac{dC}{dx}$ in the previous equation an expression is obtained containing only S and x as variables. Integrating this twice, S is found, and C can then also be determined for any point, as well as the deflections and inclinations.

The same operation has to be carried out with a uniformly distributed pressure p, and also with an unperforated plate; but the resultant formulæ, as already mentioned, are too long to be reproduced here, and have not been practically applied. See Grashoff, 1866, pp. 248, 254, 263.

A fair idea as to what happens in stayed plates may be arrived at by examining the so-called **continuous beams**, which, like railway rails, are supported at more than two points. The analysis is carried out as follows :—

Each span (fig. 145) should be examined separately. There will be the uniformly distributed pressure p, and the supporting pressures $Q_1+Q_2=p\cdot l$. Then there will be the two external moments m_1 and m_2, and the deflections a_1 and a_2, and the depression y_2. Any of these values may be zero, in which case they are struck out of the fol-

lowing two equations, obtained by integrating the elastic line

$$\frac{d^2y}{dx^2}=\frac{m}{E.I}.$$

$$a_2=a_1+\frac{m_2 . l}{E.I}-\frac{Q_2 . l^2}{2.E.I}+\frac{p . l^3}{3.E.I}.$$

$$y_2=a_1 l+\frac{m_2 . l^2}{2.E.I}-\frac{Q_2 . l^3}{3.E.I}+\frac{p . l^4}{8.E.I}.$$

Instead of the value $\frac{1}{E}$ it will be necessary to write $\frac{1}{E}\left(1-\frac{1}{\mu^2}\right)$ if the plate remains flat across its width. Applied to stayed plates this

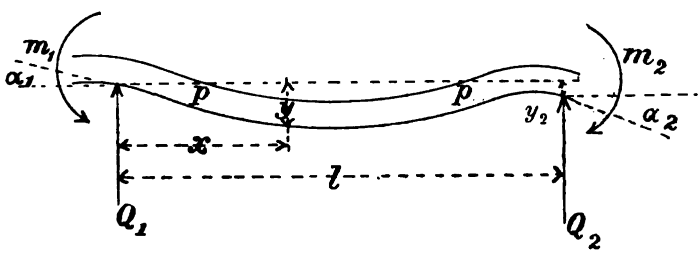


FIG. 145.

correction would give the approximate mean deflection and depression from stay to stay.

Having obtained the above equations for each span, these should be resolved, then Q, m, a and y can be found.

If the supports are numerous

$$Q_1=Q_2=\frac{p . l}{2},$$

$$m_1=m_2=\frac{p . l^2}{12}, \text{ and } S=\frac{p . l^2}{2 . t^2}.$$

At the centre, between two supports, the bending moment is $m_0=\frac{p . l^2}{24}$.

For stayed flat plates these are only mean values of the moments which are distributed over the breadth A. If the deflections y_1 and y_2 (fig. 146) should be found to be as one to two, it would be fair to as-

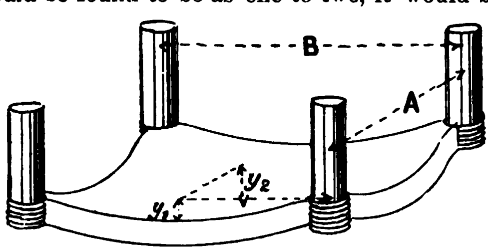


FIG. 146.

sume that the moments are uniformly distributed over the width; but a rough estimate shows that y_2 is only about $1\frac{1}{3} . y_1$, so that m_1 would

be about six times greater than m_0, from which it follows, that were the plate not perforated by the stay holes, the radial stresses at these points would be about

$$S = \frac{3 \cdot p \cdot l^2}{2 \cdot t^2}, \text{ instead of } \frac{p \cdot l^2}{2 \cdot t^2}, \text{ as for a continuous beam.}$$

The cross stresses near the stay holes combine and form circumferential stresses, similar to those which exist in thick-walled cylinders, and must be greater than the above value, so that the limit of elasticity of the metal, close to a stay with a riveted head, will have been nearly reached at double the working pressure for boilers, as now constructed. Of course when nuts and washers are fitted the case is materially altered.

Irregularly Stayed Plates.—By applying the formulæ of continuous beams to irregularly stayed plates—for instance, such as occur in boiler backs (fig. 147)—it can be shown that the stresses near the stays on

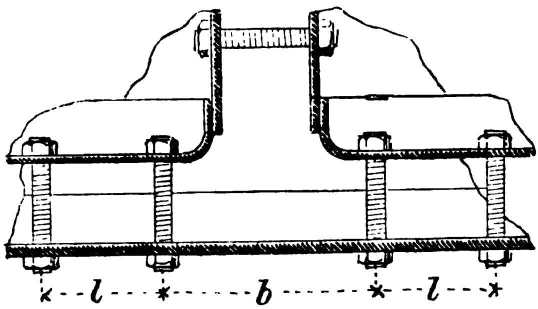


FIG. 147.

either side of the pitch b are greater than the usually accepted formulæ would lead one to expect. The value of m_1 has to be increased from

$$\frac{p \cdot l^2}{12}, \text{ to } \frac{p \cdot b^3}{l \cdot 12}.$$

It is exceedingly difficult to arrive at a satisfactory view of the case illustrated in fig. 148, where the pitches A and B are not equal. The stresses set up parallel to the pitches A will certainly be affected by the pitch B, for the more this is reduced the less metal will remain between the stays, and in the extreme case when they nearly touch each other, i.e. when $B = d$, the stress S_1, due to the curvature at the line of holes, would be increased to $\frac{2 \cdot p \cdot l^3}{d \cdot t^2}$; and comparing this with the above value S, for such cases when A and B are equal, we find that $\frac{S_1}{S} = \frac{4}{3} \cdot \frac{A}{d}$.

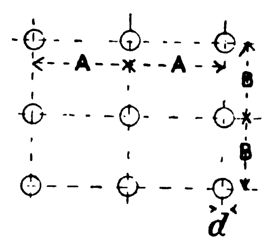


FIG. 148.

On the other hand, when A and B are nearly equal, a slight reduc-

tion of B will tend towards a more uniform distribution of the stresses between the stays, thereby reducing the maximum values of S ; and it is possible that an alteration of B may in the one case reduce, and in the other case increase, the maximum stresses.

Diagonal Pitches.—The Board of Trade has adopted the view that the maximum stresses are proportional to the product of A into B, but certain allowances are made for large variations (see p. 324). Lloyd's Register, on the other hand, does not take into account the smaller of the two pitches. It has been suggested that the maximum stress can be best expressed as being proportional to the sum of the squares of the two pitches. This amounts to the same thing as substituting the diagonal for the square pitch, and multiplying the constants of the various formulæ by two.

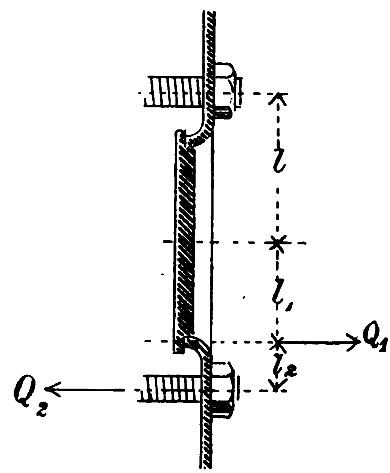


FIG. 149.

Manholes.—A problem which presents itself with reference to the fitting of manholes in flat plates is the following (see fig. 149) :—

Let $Q_1 = p \cdot l_1$, where p is the working pressure and l_1 is half the width of the manhole door ; then the bending moment at the position of the stay is

$$m = p \cdot l_1 \cdot l_2 + \frac{p \cdot l_2^2}{2}.$$

If experience has shown that for this particular thickness of plate a pitch of stays equal to L is permissible, then evidently the following equation must be true :—

$$\text{(See p. 137.)} \quad \frac{p \cdot L^2}{12} = p \cdot \left(l_1 \cdot l_2 + \frac{l_2^2}{2} \right).$$

So that the maximum value for $l = l_1 + l_2$ is

$$l = l_1 \sqrt{1 + \frac{1}{6} \cdot \frac{L^2}{l_1^2}}.$$

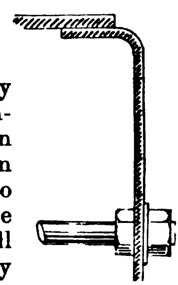


FIG. 150.

Effects of Boiler Deformations.—A little familiarity with the working of the equations for a_2 and y of continuous beams (see p. 137) will soon lead to the conviction that the changes of form to be expected in boilers often produce stresses which far exceed any that may be due to the steam pressure alone. As a simple instance take the case of a steam-space stay placed too close to the shell plate of a long double-ended boiler (fig. 150). Its duty is to support the surrounding flat plate ; but, on account of the longitudinal contraction of the shell, the stay will be in compression and actually assist the steam pressure in forcing out the plate, thereby seriously increasing the load on the adjoining stays, and increasing the bending stresses in the flange.

Screwed Stays near Flanges.—The reverse action is met with at the edges of the combustion chamber backs. Evidently there is a pull in one direction at the flange, which is balanced by the pull on the stay (see fig. 151). This load Q_1 is proportional to the mean of the pitches l_2 and l_1, and if the latter dimension is the maximum which

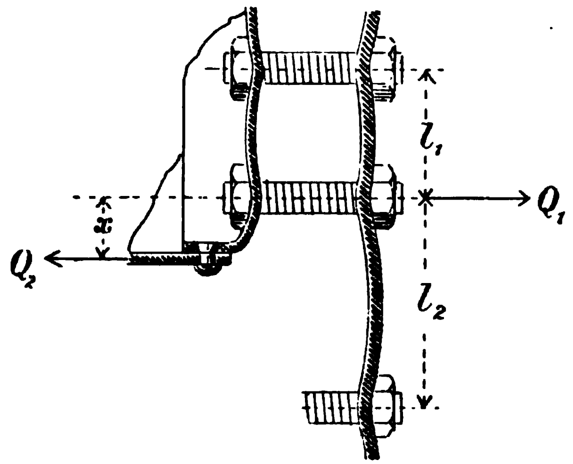


FIG. 151.

practice has shown to be safe, then the distance x, measured from the centre of the flange, is found by the formula

$$x = -\frac{a}{2} + \sqrt{\frac{a^2}{4} + \frac{l_1^2}{6}},$$

where a is the width of the water-space between the two combustion chambers.

The following list contains most of the experiments that have been made on flat plates :—

W. Fairbairn, 1856, pp. 194 and 197. $\frac{1}{2}$-in. copper and $\frac{3}{8}$-in. iron plates, 5-in. and 4-in. pitches.

'Franklin Inst.,' 1872, iii. vol. lxiii. p. 93. $\frac{5}{16}$-in. iron plate, $8\frac{3}{4}$- and $9\frac{3}{16}$-in. pitch.

L. E. Fletcher, 'Rep. M. S. U. A.,' Oct. 1876, p. 31. Admiralty experiments on 'Thunderer' boiler.

R. Wilson, 'Enging.,' 1877, vol. xxiv. p. 239. Five experiments on unstayed boiler ends.

W. Boyd, 'M. E.,' 1878, p. 223. $\frac{1}{2}$-in. iron, $\frac{7}{16}$-in. steel plates, 9-in. pitch.

D. Greig and Max Eyth, 'M. E.,' 1879, p. 272, $\frac{3}{8}$-in. and $\frac{9}{16}$-in. plates, $4\frac{1}{2}$-in. pitch.

W. S. Hutton, p. 180.

Board of Trade experiments on mild steel, 'Parl. Rep.,' 1881 (c. 2897), 1885 (c. 4572).

Of these experiments the last are undoubtedly the most important, and as the reports contain the necessary drawings, they might be used

for a more thorough analysis of the subjects than has yet been attempted, and a short summary of the results is therefore reproduced here. As the maximum pressures are of less interest than those under which the first permanent set was noticed (p_1), when the scale fell off the plates near the stays (p_2), and when cracking noises were first heard (p_3), only these pressures have been noted. See following table.

Board of Trade Experiments on Flat Plates.

Plate Thickness	Stays			Pressures and Deflections at Centre					
	Diam.	Pitch	Mode of Attachment of Stays	p_1	Set	p_2	p_3	Bulge	Set
Inch	Inches	Inches		Lbs.	Inch	Lbs.	Lbs.	Inch	Inch
$\frac{9}{32}$	$1\frac{1}{2}$	8	Riveted	200	·005	300	...	·42	·35
,,	$1\frac{1}{2}$	8	Nutted	200	·03	350	...	·25	·19
$\frac{5}{16}$	$1\frac{1}{2}$	$11\frac{5}{16}$	Riveted	100	·24	100	130	?	?
,,	$1\frac{1}{2}$	$11\frac{5}{16}$	Nutted	75	·01	...	...	...	...
$\frac{13}{32}$	$1\frac{1}{2}$	8	Riveted	325	·005	...	375	·07	·025
,,	$1\frac{1}{2}$	8	Nutted	300	·01	550	...	·32	·29
,,	2	8	,,	400	·01	500	500	·26	·22
$\frac{7}{16}$	$1\frac{1}{2}$	$11\frac{5}{16}$	Riveted	150	·01	...	...	...	...
,,	$1\frac{3}{4}$	$11\frac{5}{16}$	Nutted	200	·03	...	...	...	...
$\frac{1}{2}$	$1\frac{1}{2}$	8	Riveted	550	·005	...	540	·045	·005
,,	$1\frac{1}{2}$	8	Nutted	550	·01	...	...	...	...
$\frac{15}{32}$	2	8	,,	500	·03	700	...	·44	·42
$\frac{1}{2}$	2	8	,,	650	·03	...	...	...	...
$\frac{9}{16}$	$1\frac{1}{2}$	$11\frac{5}{16}$	Riveted	225	·01	...	...	...	...
,,	2	$11\frac{5}{16}$	Nutted	200	·015	350	...	·16	·11
,,	$2\frac{1}{4}$	$14 \times 15\frac{1}{2}$	Two nuts	150	·01	...	...	...	...
,,	$2\frac{1}{4}$	$14 \times 15\frac{1}{2}$	,,	175	·02	...	...	...	...
,,	$2\frac{1}{4}$	14×15	Nuts and washers	300	·03	...	...	...	...
,,	$2\frac{1}{4}$	14×15	Nuts and washers	250	·01	...	...	...	...

From these results it would appear that the diameter of the stay, or of its nut or washer, is of as much importance, if not more, than the thickness of the plate, the maximum stresses, as shown by the falling off of the scale, being found near the stays. As this phenomenon is only noticed with tensile test pieces when the plastic limit is reached, and the load is about 50 % of what they can stand, very serious stresses are evidently set up in flat plates at comparatively low pressures.

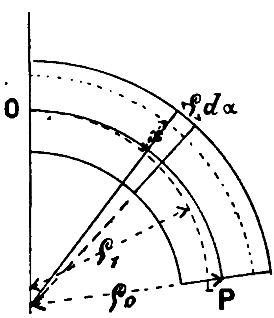


FIG. 152.

Curved Beams.—An originally curved beam behaves very much in the same way as a straight one, except that the stresses are not distributed so uniformly. Let the line OP (fig. 152) be the neutral fibre, which does not alter its length while being bent. For any short length $\rho \,.\, d\alpha$ the elongation of the fibres above or below this line is proportional to their distances from it. But not so the strains, for, as the outer fibres are longer than the inner ones, so will the strains, and consequently also the stresses, be proportionately greater the nearer they are to the centre of

curvature. Let ρ_0 and ρ be the radii of curvature of the neutral fibre before and after bending, and let x be the distance of any fibre from this line ; then, if E is the modulus of elasticity, it is easily shown that the stress is

$$S = E \cdot \frac{x \cdot (\rho_0 - \rho)}{(x + \rho_0) \cdot \rho}.$$

Then, on carrying out the necessary algebraical operations, n (figs.

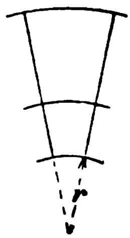


FIG. 153.

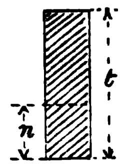


FIG. 154.

153 and 154), the distance of the neutral fibre from the concave side of the rectangular beam, is found.

$$n = t\left(\frac{1}{\log \text{nat} (1 + m)} - \frac{1}{m}\right).$$

In this formula $m = \frac{t}{r}$.

The following are a few numerical values :

$m =$	∞	10	5	1	$\frac{1}{2}$	$\frac{1}{10}$	0.
$\frac{n}{t} =$	0	·317	·358	·443	·466	·492	·500.

The maximum stress S exists on the concave side of the beam.

$$S = C \cdot 6 \cdot \frac{M}{bt^2}.$$

In this formula M=bending moment and

$$C = \frac{1}{3} \cdot \frac{m\,[m - \log \text{nat} (1 + m)]}{(m + 2) \log \text{nat} (1 + m) - 1}$$

The following are a few numerical values:

$m =$	∞	10	5	1	$\frac{1}{2}$	$\frac{1}{10}$	0.
$C =$	∞	2·889	2·103	1·287	1·154	1·033	1·000.

The practical conclusions to be drawn from these figures are that the bending stresses on the concave side of plates, flanged to the usual radius of, say, twice the thickness, are 15 % greater than those in the adjoining flat plates ; that when the inner radius is reduced to the thickness of the plate 29 % has to be added to the stress, while when there is no round on the inside of the flange the stresses are infinitely great. The end of every crack can also be looked upon as the inner

surface of a curved beam, and the lightest external force would extend it, unless there are other stresses beyond the crack holding the plate together. When they are absent, or when tension stresses exist in the plate, there is nothing to stop the crack from extending with the usual explosive rapidity. Similarly, the concave sides of cold bend samples are very weak, on account of their small radii, and, as the residual stresses in the samples all tend to produce tensions at this point, it is not surprising that the majority of such test pieces crack as shown in fig. 155 some time after they have been put aside, and without any apparent cause.

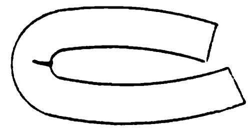


FIG. 155.

Elastic Deformation of Curved Beams.—This subject is not of such importance as to warrant the development of the unavoidably long formulæ, but for those who wish to study the behaviour of flanged end plates or furnaces it is well to point out that there are two motions—a and b in figs. 156, 157. In the one case it is assumed that the entire flange is tilted through an angle α, and in the other a uniformly

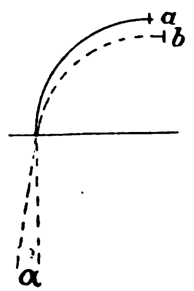


FIG. 156.

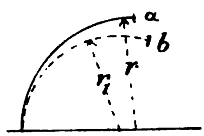


FIG. 157.

distributed bending moment is supposed to have reduced the radius r to r_1. In the one case $a = b = r \cdot \alpha$, in the other $a = r - r_1$, and $b = (r - r_1)\left(\frac{\pi}{2} - 1\right)$.

Shearing Stresses in Beams.—No shearing stresses exist in a bent beam when it is exposed to only two external bending moments, m, m (fig. 158). In every other case the shearing force in any particular section of a beam is exactly equal to the otherwise unbalanced load at that point. Thus, in fig. 138, p. 132, the shearing force at the point x is $Q - p\ (l - x)$.

FIG. 158.

If this force were evenly distributed over the sectional area of the beam, it would be easy to calculate it; but there are certain conditions which have to be fulfilled, and which lead to somewhat complicated results, particularly if the section is an irregular one. In rectangular beams, and therefore also in flat plates, the case is simple.

Let σ represent the intensity of the shearing stress at a point situated at the height y above the neutral fibre op of a beam

(fig. 159), while S is the longitudinal stress at that point; then it is easily proved that

$$\frac{d\sigma}{dy} = -\frac{dS}{dx}.$$

But, as $\frac{dS}{dx} = \frac{12}{t^3} \cdot y \cdot \frac{dm}{dx}$, we have

$$\frac{d\sigma}{dy} = C \cdot y, \text{ where C is a constant.}$$

If, then, the total shearing force in the section at the distance x is Σ, we have

$$\sigma = \frac{3\Sigma}{2 \cdot t}\left\{1 - 4\left(\frac{y}{t}\right)^2\right\}.$$ [1]

Fig. 161 represents the distribution of the longitudinal stresses, and fig. 160 that of the shearing stresses.

As a shearing stress σ is a compound of a tension and compression

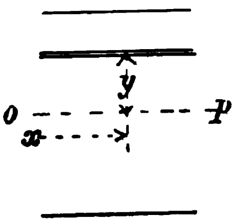


FIG. 159.

FIG. 160.

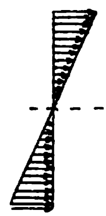


FIG. 161.

stress S, acting at angles of 45° to the line of shear, as shown in fig. 125 (p. 128), it is easily proved that

$$\sigma = \pm S.$$

In order, then, to resolve the various stresses to be met with in a beam into their respective right-angled resultants it is only necessary to apply the formulæ of p. 132; then

$$tg \cdot 2 \cdot \gamma_1 = \frac{o + o + \sigma + \sigma}{-p + S + o + o} = \frac{2 \cdot \sigma}{S - p}.$$

$$x = \frac{p+S}{2} + \sqrt{\frac{4 \cdot \sigma^2 + (S-p)^2}{2}}, \quad y = \frac{p+S}{2} - \sqrt{\frac{4 \cdot \sigma^2 + (S-p)^2}{2}}.$$

Here p is the transverse pressure, but the signs of both S and p will have to be changed if they are compression stresses. The distribution of p is determined by σ.

It is now possible to construct curves showing the direction and intensities of the resultant stresses at any point of a beam (see p. 162).

Plastic Beams.—When flanging or bending plates, the elastic limit of their material is always passed, and the resultant deformation is a permanent one. In the chapter on 'Strength of Materials' a formula has been explained which enables one to calculate the stresses in the

[1] The axial shearing stresses on the threads of a screwed stay can be shown to be distributed in the same manner, and instead of making the thickness of the plate or nut half the diameter, it ought to be three-quarters.

outside fibres of narrow plastic beams, when their curvature is known. Roughly $S = \frac{4 \cdot m}{t^2}$; so that S is only two-thirds of what it would be in an elastic beam.

Another very important difference between an elastic and a plastic beam is that the stresses of the former bear some relation to the deformation, while those of the latter are nearly, if not quite, independent of the same.

The work W_1 required to bend an elastic steel beam is proportional to the square of its final curvature, about

$$W_1 = \frac{280 \cdot A \cdot t^3}{\rho^2} \text{ foot tons.}$$

Here A is the superficial area of the plate, measured in square feet, and ρ the radius of curvature, measured in feet. By the time that the elastic limit is reached this amounts to about

$$W_1 = \frac{A \cdot t}{450} \text{ foot tons,}$$

and is independent of the radius. In other words, the maximum energy which can be stored in a flat spring is proportional to its volume.

The work W_2 required for bending a plastic beam or plate is proportional to the curvature.

$$W_2 = \frac{C \cdot A \cdot t^2}{\rho}.$$

The value of the constant C is about

6	for mild steel, cold.	5	for wrought iron, cold.
4	„ „ „ blue hot.	$3\frac{1}{2}$	„ „ „ blue hot.
0·3	„ „ „ red hot.	0·25	„ „ „ red hot.

Thus to bend a cold steel shell plate, whose dimensions are 14 feet × 7 feet × 1 inch, and whose radius is 6·7 feet, a power of 88 foot tons would be required. To do it at a red heat requires only $4\frac{1}{2}$ foot tons.

The above values have been estimated from the following experiments and other observations. Dr. J. Kollmann ('Ver. Gew.,' 1880, vol. lix. p. 107, &c.) states that he found the limits of elasticity (? plasticity) to be as follows :—

Tables of Limits of Elasticity or Plasticity.

Temperatures, ° F.	68°	1,380°	1,470°	1,560°	1,650°	1,750°
Iron . . . Tons	17·5	2·0	1·3	1·0	...	...
Mild steel . „	25·4	3·0	...	..	1·5	1·2

C. E. Stromeyer, 'C. E.,' 1886, vol. lxxxiv. p. 125, Nos. 4, 9, 10, 12, 18.

Temperatures, ° F. Colour of Fracture	68° ...	470° Straw	500° Blue
Hard steel . . . Tons	19·4	17·0	15·9
Mild „ . . . . „	16·3	...	10·0

From this it will be seen that at a blue heat and at a red heat iron, as well as steel, grows more pliable in the ratio respectively of 3 to 2 and 20 to 1. The power required to permanently bend an iron and a steel plate 12 inches wide and 1 inch thick is, therefore, as follows :—

Condition	Cold	Blue Hot	Red Hot
Iron plate . . Foot tons	60	40	3
Steel „ . . „ „	75	50	3¾

Cross Deformation of Plastic Beams.—When the elastic limit has been passed the value of $\frac{1}{\mu}$, the coefficient of cross contraction, increases from $\frac{1}{3}$ to $\frac{1}{2}$. For very narrow beams the ratio of ρ to ρ_1 (fig. 162) would then be as 1 to 2. In wide ones, such as shell plates, this deformation is of course impossible, and it is evident that a cross stress must have been set up, whose intensity is nearly equal to half the longitudinal one, or, say, + 12 tons per square inch under the action of the bending rolls. But this condition also demands supplementary longitudinal stresses at the edges of the plate and compression stresses near its centre. They are all modified by the spring of the plate as it leaves the rolls ; but it is quite clear that the residue is both of a severe and of a complicated nature. Stresses of a similarly complex nature also exist in small bent test samples ; and it is interesting to note that when the steel is of an unsatisfactory quality it usually cracks at the centre of the width, showing that it was incapable of withstanding the combination called drum tension (see fig. 163 and p. 128) to be found there.

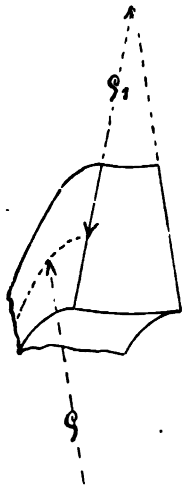


FIG. 162.

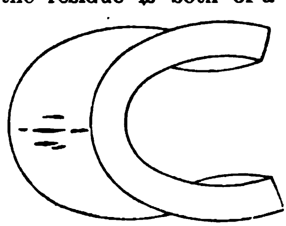


FIG. 163.

The complicated nature of these stresses, particularly in wide beams, readily explains that the maximum stresses to be found in them when the plastic limit has once been passed differ very materially from those found by the formula

$$S = \frac{6 \cdot m}{t^2}.$$

Shearing Stresses in Plastic Beams.—The previously found formula, $\frac{d\sigma}{dy} = -\frac{dS}{dx}$, is independent of the elasticity of the material. If, therefore, the longitudinal stresses in a rectangular plastic beam are distributed as shown in fig. 164, then the shearing stresses must be distributed as in fig. 165. This illustration represents a case where the bending stresses have been produced by means of a light force acting on a long leverage, as, for instance, when steel plates are being passed through the bending rolls.

The case is very different when a strong force with a short leverage produces the same moment. Then the shearing stresses will be nearly uniformly distributed over the thickness (fig. 167), and the longitudinal stresses will have to accommodate themselves accordingly somewhat after the fashion shown by the curves in fig. 166. In the one case the beam gives way at the outside fibres, in the other at the neutral one.

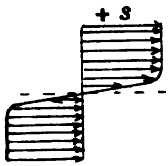


FIG. 164.

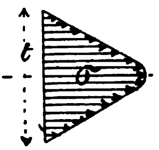


FIG. 165.

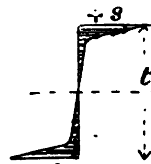


FIG. 166.

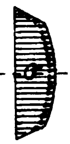


FIG. 167.

An intermediate stage is worthy of mention. Suppose that the plastic limit of tension and compression is 20 tons, while for shearing it is 15 tons; then both outside fibres of a beam would give way to longitudinal stresses, at the same time that the centre fibres commence to shear, if these stresses are reacted on simultaneously. This happens in a square bar if the bending force is applied at a distance equal to $\frac{1}{3}$ of its thickness, and doubtless most rivets behave in this way.

Stresses in Cylindrical Shells.—The circumferential stress in a cylindrical shell of the mean diameter D is found by the well-known formula

$$S = \frac{p \cdot D}{2 \cdot t},$$

where p is the internal pressure and t the thickness of the shell plate. If the ends of the shell plates are joined together by riveted longitudinal seams, the percentage of their strength will have to be taken into account. If the boiler shell is built up of several strakes, and the longitudinal seams break joint, it has been argued that the stresses are proportionally reduced, and that they should be calculated as follows:

$$S = \frac{p \cdot D \cdot l}{2 \cdot t \cdot \Sigma(\Delta l)}.$$

Here l is the length of the boiler, and $\Sigma(\Delta l)$ is the sum of the widths of the plates, including the flanged end plates, from which the rivet holes of only one longitudinal seam have been subtracted. As $\Sigma(\Delta l)$ is often greater than l, this view would lead to the conclusion that a riveted shell is stronger than a solid one. However, this argument is only true as regards the mean stresses, and that it leads to valueless results is proved by the fact that the same reasoning applied to a beam would lead to a wrong conclusion, for its mean stress is just nothing, being + S at the top and − S at the bottom.

A point where variation of stresses in shell plates may be expected is at A (fig. 168), in the solid plate near the end of a longitudinal seam. Let it be assumed that this seam is more elastic than the plate, i.e. that whereas a stress of one ton would stretch the latter $\frac{1}{13000}$ of its length, the same stress would cause the joint to spring or stretch, say,

four times as much. Then, as these two parts are firmly connected by the circumferential seam, the solid plate at A would be subjected to a four times greater stress than the joint. A solitary and perhaps not very reliable experiment on this subject showed that the stress was actually eight times greater, and similar results were obtained near the single butt-strap joints on ships' sides (C. E. Stromeyer, 'N. A.,' 1886, vol. xxvii. p. 34).

The remedy which readily suggests itself is to make the longitudinal seams more substantial—say, of the double butt-strap type. Now, however, this part may be more rigid than the plate, and will have to bear a proportionally heavier load, and being perforated is less capable of sustaining it. A more correct principle would be to make the longitudinal joints exactly as elastic as the solid plate. A few remarks on the elasticity of a riveted joint will be found further on (p. 168), when discussing their theories, but a true solution can only be obtained by careful experiments.

The order in which seams naturally range themselves as regards rigidity is—

1st. Single butt-strap joint.
2nd. Lap joint.
3rd. Butt joint with one wide and one narrow strap.
4th. Butt joint with two equally wide straps.

Of course the number of rows of rivets, their pitch and diameter, and the thickness of the cover plates, will affect the results.

The Longitudinal Contraction of Cylindrical Shells is $\frac{S \cdot l}{E \cdot \mu}$, where l is the length and S the circumferential stress. In a boiler of 17 ft. length this will amount to about 0·025 in. at the ordinary working pressure. If no stays are fitted to take up the longitudinal stress, which is one-half of the above, it will be found that the elongation, $\Delta l = \frac{S}{E} \cdot l \left(\frac{1}{2} - \frac{1}{\mu}\right) = \frac{p \cdot D}{E \cdot T} \cdot l \left(\frac{1}{2} - \frac{1}{\mu}\right)$. This can be utilised for the determination of $\frac{1}{\mu}$, the coefficient of cross contraction. By riveting longitudinal strips on a boiler shell, the longitudinal contraction is reduced, at least locally, and this is exactly what all riveted longitudinal seams do. Consequently longitudinal compression stresses are set up in them, which must produce tension stresses in some other part of the shell, probably along either side of the seam. A longitudinal compression stress C will also be set up at A (fig. 168), which, combining with the circumferential stress S, constitutes a shearing stress, and that, as has been shown by experiments (see fig. 123, p. 126), is far more injurious to iron and steel than a simple tension. Therefore, whether elastic lap joints or the more solid butt-strapped joints are used, very severe stresses will be found in the adjoining plates.

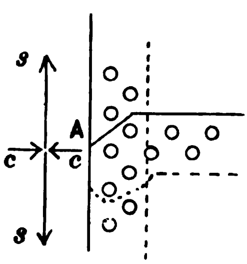


FIG. 168.

Influence of End Plates on the Stresses in Boiler Shells.—In fig. 169 let the line CD represent the shape of half the length l of a boiler shell of the diameter D, while subjected to an internal pressure p_1. The shell plate will acquire this curvature if secured to the end plate at C, as shown; if unsecured it would have remained cylindrical and

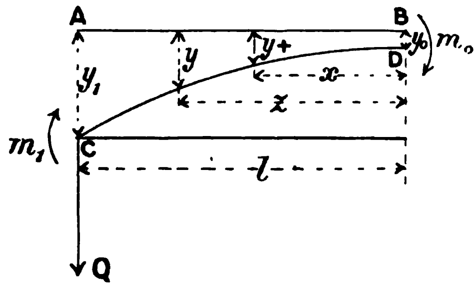


FIG. 169.

have occupied the position at the line AB. Under that condition the increase of half the diameter is $y_1 = \frac{p_1 \cdot D^2}{4 \cdot E \cdot t}$, where t is the thickness of the plate and E the modulus of elasticity. The stress will be $S_1 = \frac{p_1 \cdot D}{2 \cdot t}$; but, as will be seen in the diagram, if the ends are held down there is no circumferential straining of the shell at C, while at D it is reduced in the ratio of $y_1 - y_o$ to y_1. The object of this investigation is to ascertain what this value is.

A little reflection will show that the circumferential stress S at any point x or z is proportional to the distance of that point of the shell from its original position.

$$S = S_1 \cdot \frac{y_1 - y}{y_1} = \frac{p_1 \cdot D}{2 \cdot t} \cdot \frac{(y_1 - y)}{y_1}.$$

But as this stress is only capable of balancing part of the internal pressure, there remains

$$p = p_1 \frac{y}{y_1},$$

which has to be transmitted to the end plate, and in doing so longitudinal bending stresses are set up in the shell, which have to be ascertained. This can be done by examining a long strip of the boiler shell—say, 1 in. wide—loaded irregularly with a pressure p, and supported at its ends by loads Q. A bending moment m_1 will also be found there.

Evidently the load Q is proportional to the area ABCD.

$$Q = \frac{p_1}{y_1} \cdot \int_o^l y \, dz.$$

The bending moment at the distance x is

$$m = -m_1 + Q(l - x) - \frac{p_1}{y_1} \cdot \int_x^l y \cdot (z - x) dz,$$

which may be written

$$\frac{E \cdot t^3}{12} \cdot \frac{d^2y}{dx^2} = m$$

$$= - m_1 + \frac{p_1}{y_1}\left[l \int_0^l y \, . dz - x \int_0^x y \, . dz - \int_x^l y \, . \, z \, . \, dz \right]$$

This leads to the equation

$$y = \frac{y_0}{2} \left(e^{ax} + e^{-ax}\right) \cos ax + A\left(e^{ax} - e^{-ax}\right) \sin ax.$$

By integration and differentiation the values of $\frac{y_0}{2}$, A, and a can be found, and numerical values obtained, when the conditions of the external forces are known. Thus, when very thick shell plates are attached to thin end plates, or when these have a very weak and well-rounded flange, $m_1 = 0$ or nearly so. In the latter of these two cases y_1 is also reduced, on account of the spring of the flange, which is proportional to Q, and this value has therefore also to be reduced, and the relief afforded to the shell plate is small. The calculations are too complicated to be reproduced here. The problem has been discussed on somewhat different lines by Dr. F. Grashoff, 1878, p. 316, and by J. T. Nicolson, 'N.-E. C. I.,' 1891, vol. vii. p. 205. Adapting some of his results to a boiler of 15 ft. diameter, with 1-in. shell plates at 100 lbs. pressure, we find the circumferential stress in the centre of the length to be as follows :

Length between end plates (feet)	∝	15	10	5	2	1
Circumferential stress at centre (tons)	3·35	3·35	3·34	3·15	2·96	·70

But this is only true if the end plates are rigid while $m_1 = 0$. In practice this is never the case, and a very considerable deformation must take place in the end plates. The back end, which is practically a flat plate, will be strained uniformly. If made exceptionally thin, it would have to expand as much as the shell, but not being able to contract crossways, the consequent drum tension would be equal to

$$\frac{p \, . \, D}{2 \, . \, t}\left(\frac{\mu}{\mu - 1}\right),$$

or about 50 % more than the circumferential tension in the shell.

With the front plate, which is perforated by furnaces, manholes, and tube holes, it would at first sight appear as if locally the stresses might grow to be excessive, but it is clear that only under exceptional circumstances could they exceed the circumferential stresses in the shell plates. The weakest points are undoubtedly to be found between the furnace front holes ; but, as no fractures have ever taken place there, it is but reasonable to suppose that the stresses in the end plates, due to their attachment to the shell, are small, the roundness of the flange providing the necessary springiness.

Stresses near the Dome Holes.—A problem which is often met with in boiler designing is the efficient staying of the corners of two

intersecting cylindrical shells (fig. 171). Fig. 170 is a section through the line $c\epsilon\delta$.

Assuming that the larger cylinder is cut up into numerous rings of the thickness δ, and that internal pressure is p, then if the ring under

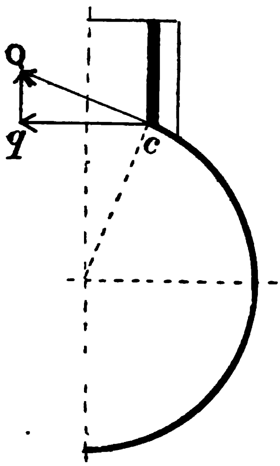


FIG. 170.

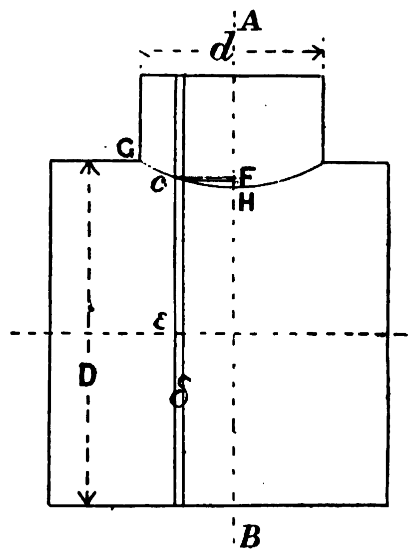


FIG. 171.

consideration were severed at the point C it would have to be retained in its position by a pull $Q \doteq \frac{D}{2} \cdot \delta \cdot p$.

Now the vertical component of this pull could be transmitted to the straight side of the cylinder d, but a stay would have to be fitted to take up the horizontal component q. A little consideration will show that it bears the same relation to Q as the distance CE bears to half the diameter D. The same arguments show that the horizontal component of the narrow strip CF is also proportional to its projected area. This shows that the greatest pull is found at G and the smallest at H, and instead of fitting the chief stays at H, as is often proposed, they should be fitted at C and G, unless this latter point is well supported by the flanges.

Stresses in Cylindrical Flues.—The same formula as for cylindrical shells may be applied here, but when experimenting, buckles will show themselves long before even the limit of elasticity for compression has been reached, and the pressure cannot then be increased, because the flue grows weaker and weaker the more it alters its shape. It is important to know at what pressure this change takes place. Let the elliptical dotted line (fig. 172) be the shape of the originally cylindrical flue, and let the black line be the curve of thrusts. This is a nearly circular line, and can be found by graphic construction, as is done in the case of arches, &c. The intensity of the thrust is $\frac{D}{2} p$, where p is

the external pressure. To find the bending moment at any position (say, at the angle a) it is only necessary to multiply this thrust into

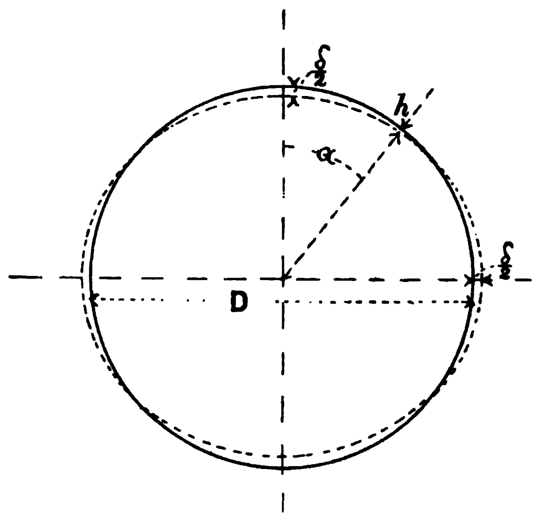


FIG. 172.

the distance h between the two lines. At the major and minor axes the moments are maxima $m_1 = - m_2 = \frac{D \cdot p \cdot \delta}{2 \cdot 2}$. For any other point

$$m = m_1 \cos 2a.$$

This equation is practically identical with that obtained for a strut of the length $\frac{D\pi}{4}$ with loose ends and loaded with the above force. If accidentally bent, such a strut would straighten itself while supporting its load, provided it does not exceed

$$K = \frac{16 . E . I}{D^2}.$$

Substituting the value for K, and also for $I = \frac{t^3}{12}$, the moment of inertia, we have

$$\frac{D}{2} p = \frac{16 . E . t^3}{12 . D^2},$$

from which it follows that p is limited:

$$p \leqq \frac{8}{3} . E . \frac{t^3}{D^3} = 8 . 10^7 . \frac{t^3}{D^3} \text{ lbs. per sq. in.}$$

As will be seen, p in this formula only depends on the modulus of elasticity, and not on the working strength of the material; or, in other words, the strength of a flue to resist a collapsing or buckling pressure depends on its rigidity, whereas its strength to resist a crushing

pressure depends on the strength of the material. The two values are equal when

$$p = \frac{2 \,.\, t \,.\, S}{D} = \frac{8}{3} \,.\, E \,.\, \frac{t^3}{D^2};$$

that is, when

$$\frac{S}{E} = \frac{4}{3} \cdot \frac{t^2}{D^2}.$$

Here S is the plastic limit of the material. This formula expresses the conditions under which a flue would give way simultaneously, both by crushing and collapsing. Assuming that $E \div S = 1200$, we have

$$\frac{t}{D} = \sqrt{\frac{3}{4800}} = \frac{1}{40}.$$

This means, that as long as the ratio of t to D is less than $\frac{1}{40}$, a perfectly cylindrical tube, unsupported at its ends, will give way by crushing and not by collapsing. Boiler tubes are not supported at their ends, which is the same thing as if they were of infinite length; yet, on account of their relative great thickness, it is not wrong to estimate their strength by the formula

$$p = \frac{2 \,.\, t \,.\, S}{D}.$$

Corrugated Flues.—By corrugating a flue, I, the moment of inertia of one inch of its section is increased from $\frac{t^3}{12}$ to $\frac{t^3}{12} + \frac{t \,.\, h^2}{8}$, where h is the depth of the corrugations, and we have

$$p \leqq 32 \,.\, \frac{E}{D^3} \left(\frac{t^3}{12} + \frac{t \,.\, h^2}{8}\right).$$

Let $h = 1\frac{1}{2}$ in., then $\frac{t^3}{12}$ can be neglected, and we find that the crushing and collapsing pressures are equal when

$$p = \frac{r \,.\, t \,.\, S}{D} = \frac{4 \,.\, t \,.\, h^2 \,.\, E}{D^3};$$

that is, when $\frac{1 \,.\, S}{2 \,.\, E} = \frac{h^2}{D^2}$. This shows that even very slight corrugations will make ordinary boiler flues independent of their end supports (see Dr. F. Grashoff, 1866, pp. 232, 235).

Influence of Rings and Furnace Ends.—The previous formulæ take no account of the strengthening effect of rings or furnace ends. One way of dealing with this question is to regard the shell of a furnace as if it were a wide flat column, supported at its ends by two solid guides (fig. 173).

Let L be one-eighth of the furnace circumference, whose diameter is D, and let l equal half its unsupported length, while q is the circumferential thrust per inch of length.

Then, if we take a narrow vertical strip (fig. 174), the bending moment would be $q \,.\, y$. This would produce a curvature

$$\frac{1}{\rho} = \frac{d^2y}{dx^2} = \frac{q \,.\, y}{E \,.\, I}.$$

If, on the other hand, we divide the shell into horizontal strips, they will be bent as shown in fig. 175. The external forces producing this

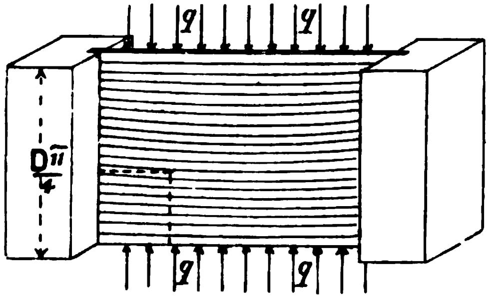


FIG. 173.

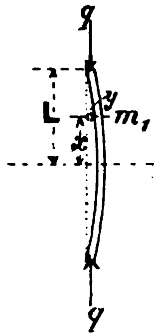


FIG. 174.

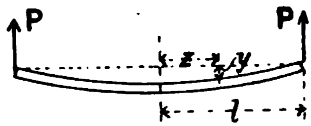


FIG. 175.

deflection are due to certain horizontal shearing forces, which when summed up can be represented by an imaginary horizontal pressure p, which is balanced by P at either end. The slight extra relief due to the action of force q on the radius ρ may be neglected (fig. 176).

We then have

$$m_1 = E \cdot I \frac{d^2y}{dx_1^2} = q \cdot y - \int_{x_1}^{L} p\,(x - x_1)\,dx + (L - x_1)\int_{x_1}^{L} p \cdot dx,$$

$$m_2 = E \cdot I \frac{d^2y}{dz_1^2} = (l - z_1)\int_{z_1}^{l} p \cdot dz - \int_{z_1}^{l} p\,(z - z_1)\,dz.$$

It is evident that to work out these formulæ would require more space than can be spared, and as the result, like the previous ones, is only applicable to perfectly circular furnaces; it will not be carried further. In practice the problem is made more complicated by the back end being irregularly supported. This is also shown in fig. 173, which represents the case of a furnace whose saddle is indicated by a vertical dotted line and is supported by the tube plate, while the bottom extends the whole length as far as the combustion chamber back plate.

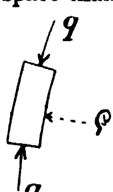


FIG. 176.

Ribbed and Flanged Furnaces.—Another way of looking at the problem, and one which is intimately mixed up with the above, is the relief afforded to the circumferential stresses by the end plates or stiffening rings. Thus in fig. 177 the line AB represents the original position of the cylindrical part of the flue, whose section is shown above. C D is its position when the diameter has decreased under pressure

if rings are fitted to the ends, while EF would be its position if no rings were fitted. The circumferential stresses are, of course, proportional to the compressions δ_1 and δ_0, showing that the relief afforded at the centre is $\frac{y_0}{\delta_1+y_1}$. In this fig. (177) y_0 and y_1 have the same meaning as in fig. 169, p. 149, and could, if required, be calculated by the formulæ to be found there. When doing this it must not be forgotten that $\frac{dy}{dx}=0$ both when $x=0$ and when $x=l$, and further that approximately

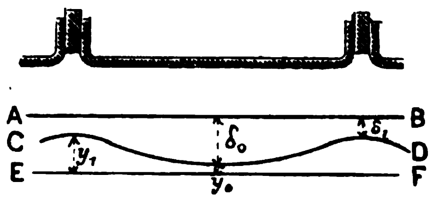


FIG. 177.

$$y_1=\frac{p_1\,.\,D^2}{4\,.\,E\,.\,t}-\frac{D\,.\,Q}{2\,.\,E\,.\,a},$$

where a is the excess sectional area of the strengthening ring over and above that of the cylindrical part, and Q has the same value as on p. 149.

Oval Furnaces.—One of the great advantages of corrugated, ribbed, flanged, and even thick-plated furnaces is that they can be made almost perfectly cylindrical, and that their strength thus depends only on the quality of the material. But none of them are ever perfect, and plain furnaces are said to exist where the excentricity amounts to 1 inch and more, and that, too, without their collapsing, at least not when worked under favourable conditions—i.e. little scale and no grease. Let δ (fig. 172, p. 152) be the difference of the largest and smallest diameter.

Then, if

$$t=\tfrac{1}{2}\text{ inch},\ \delta=\tfrac{1}{2}\text{ inch},\ D=40\text{ inches},\ p=100\text{ lbs.},\ E=30{,}000{,}000,$$

the maximum circumferential compression stress C will be

$$C=4{,}000\left\{1+\tfrac{3}{2}\,.\,\frac{30{,}000{,}000}{10{,}800{,}000}\right\}=4{,}000+22{,}222=28{,}222\text{ lbs.}$$

So that, instead of finding a compression stress of only $1\frac{3}{4}$ ton, it exceeds 12 tons, and at the same time the difference of the diameters has been increased from $\frac{1}{2}$ inch to 1·31 inch.

It must not be forgotten that the end plates are in this case of more importance to the furnace than when they are perfectly circular, but the difficulty of estimating their influence is even greater than in the last instance.

When the plates get heated and the limit of elasticity is reduced the ends are probably of little assistance, and it is under these conditions that nearly all collapses take place.

Longitudinal Stresses in Furnaces.—As in beams and in shell plates, so there will be cross stresses in furnaces, and it is of importance, particularly when testing them, to be able to ascertain their intensities. Thus in the case of the above-mentioned deformed furnace

there will be longitudinal tension and compression stresses amounting to about 4 tons per square inch, all tending to make the ends of the furnace more oval than its centre.

Perfectly circular furnaces tend to elongate under pressure, just as boiler shells contract. When testing a furnace in a shell to which it is firmly riveted at the ends, longitudinal stresses are set up. It will be found that the furnace is then subjected to a longitudinal compression stress, which is about 30 per cent. of its circumferential stress, so that under these conditions, which are not necessarily reproduced in practice, the plain furnace plates are subjected to a shrinking stress, while this is not the case with corrugated ones. Plain and also ribbed furnaces might therefore be expected to do better in a boiler than in the test tube.

This view is supported by the fact that when experimentally testing furnaces with welded seams they usually give way along a line exactly opposite the riveted seam of the shell, being the point where the longitudinal stresses would be greatest.

Theories on the Strength of Furnaces have been evolved by Euler; Fairbairn; W. C. Unwin, 'C. E.,' 1876, vol. xlvi. p. 236; Th. Belpaire, 'An. Génie,' 1879, 2nd ser. vol. viii. p. 177; W. J. Ellis, 'Enging.,' 1889, vol. xlvii. p. 19; D. K. Clark, 'Enging.,' 1889, vol. xlvii. p. 93.

A summary of experiments and theories will be found in the following Board of Trade Reports on explosions: 'Parl. Rep.,' C. 2662, April 7, 1881, May 18, 1882, Nos. 241, 257, 260, 307, 326, 346, and C. 3227 and 4573.

Experiments will also be found in W. Fairbairn, 1856, p. 1; R. V. J. Knight, 'C. E.,' 1878, vol. li. p. 133; W. H. Stock, 1880, p. 113. Of these Fairbairn's experiments are the best known and the most numerous. With the exception of his Nos. 7, 8, 10, and 11, which differ amongst themselves by about 35 %, the results agree generally with the view that the strength of a flue is inversely proportional to its superficial area; but an examination of his experiments shows that he assumed the 150 square feet of tinned iron sheeting from which he made the flues to be of a perfectly uniform thickness, whereas it is more than probable that the differences of $\frac{1}{100}$ in. must have existed, and this represents 20 % of the thickness or 35 % of the square of the thickness. It is also more than probable that the large tubes would contain more weak places than the small ones, would be less perfectly circular, and would collapse sooner than these. With the exception of these tests, in which the thickness of the plates was only ·043 in., the table on p. 157 contains the chief results of experiments on plain furnaces. For fuller details see the original papers.[1]

It would be useless to attempt the construction of a formula which would embrace these results, not only because various joints are used, but also because only in the fewest cases have measurements been taken to ascertain by how much the furnaces were oval before the test, or what was the strength of the material.

[1] All the exyeriments on patent furnaces carriod out for the Board of Trade and for Lloyd's Register recently have been made public (Morison, 'N.-E. C. I.,' 1892). They are too numerous to be reproduced here.

Experimenter or Author	Thickness	Diameter	Length	Joint	Collapse Pressure	Remarks
	Ins.	Ins.	Ins.		Lbs.	
Fairbairn . .	$\frac{1}{8}$	$14\frac{5}{8}$	60	Lap	125	...
" . .	·14	9	37	"	262	...
" . .	·14	9	37	Butt	378	...
" . .	·157	7·87	276	Lap	110	...
" . .	$\frac{1}{4}$	$18\frac{3}{4}$	61	"	420	...
Stock . .	$\frac{1}{4}$	71	54	Butt	128	$\frac{1}{2}$-in. oval
" . .	$\frac{1}{4}$ bare	71	54	"	105	
" . .	$\frac{1}{4}$	71	27	"	134	Adamson's rings
Board of Trade .	$\frac{1}{4}$	54	24	"	128	" "
Knight . .	$\frac{1}{4}$	36	24	Weld	235	...
" . .	$\frac{1}{4}$	36	48	"	217	...
Board of Trade .	$\frac{11}{32}$	$44\frac{3}{8}$	$38\frac{1}{2}$	Lap	200	Donkey boiler
" " .	$\frac{3}{8}$	43	46	"	180	" "
" " .	$\frac{3}{8}$	38	84	Weld	187·5	...
Fairbairn . .	·34	$33\frac{1}{2}$	360	Lap	99	...
" . .	$\frac{3}{8}$	42	420	"	97	...
" . .	$\frac{3}{8}$	42	300	"	127	...
Knight . .	$\frac{3}{8}$	36	24	Weld	468	...
" . .	$\frac{3}{8}$	36	48	"	390	Adamson's rings.
Board of Trade .	·42	32	94	"	370	Steel
" " .	·42	32	46	"	600	" Adamson's rings
" " .	$\frac{1}{2}$	38	86	"	450	Adamson's rings
" " .	$\frac{17}{32}$	37	108	Lap	260	Old boiler
Howden and Co. .	$\frac{17}{32}$	43	23	Weld	840	Steel, Adamson's rings

A summary of all the welded or butt-strapped plain furnaces leads to the conclusion that they gave way when the circumferential stress had reached the following values :

Thickness of plate . . . Ins.		$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{7}{16}$
Mean stress while collapsing . Tons		$5\frac{1}{2}$	$7\frac{1}{2}$	9	$7\frac{1}{2}$	15

The last of these values is for steel, and it will be seen that it is difficult to draw any conclusions as to the influence of thickness. The amount of weakness imparted to a flue, if made oval, can also not be ascertained from these experiments, but the following fact may be used as a guide. Several furnaces 3 ft. in diameter and $\frac{3}{4}$ in. thick had slowly altered their shape under a steam pressure of 180 lbs. during a period of eight months. A measurement of the furnace fronts showed these to be oval to the extent of $\frac{3}{4}$ in., while in some of the back ends the differences in the diameters amounted to $1\frac{1}{2}$ in., from which it will be seen that even under these exceptional circumstances a nominal circumferential stress of two tons per square inch can just be resisted. Assuming that the average differences of diameters at the centres of the lengths was 1 in., then an extra stress of 4 tons will have to be added to the above, showing that under the conditions of actual working, which includes high temperatures and irregular support at the back ends, mild steel is just capable of resisting a compression stress of 6 tons per square inch, which is not very far removed from the elastic limit at a temperature of 500° F. (viz. 10 tons), which may be expected to exist in furnace plates when covered with scale.

Curved Plates.—In the above case the radius of curvature of the furnace, instead of being constant (18 ins.), varied from 17 ins. at the crown to 19 ins. at the sides, and this irregularity was evidently sufficient to overstrain the material. What, it may be asked, will be the effect if the curvature changes still more rapidly, viz. from, say, 24 ins. to a straight line, as often happens in combustion chamber tops (fig. 178)? or if it changes from one side to the other, as in the case of combustion chamber bottoms (fig. 179)? When the pressure is on the convex side, the circumferential stresses are all compression, while when it acts on the concave side they are tension. This is shown in fig. 179, which represents part of the bottom of a combustion chamber with two furnaces.

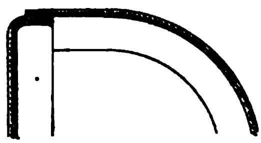


FIG. 178.

As sketched, the conditions are more unfavourable than when building an arched bridge without abutments and without any support, for the ends are in this case actually being pulled at. The stresses which are necessarily set up are of the most complicated nature. Thus there will be circumferential shearing stresses, which increase very seriously towards the forward and back ends of these plates, particularly along the line of change of flexure. Horizontal seams should not be fitted here. These shearing stresses are also very severe on the circumferential seams of the furnace back ends and of the combustion

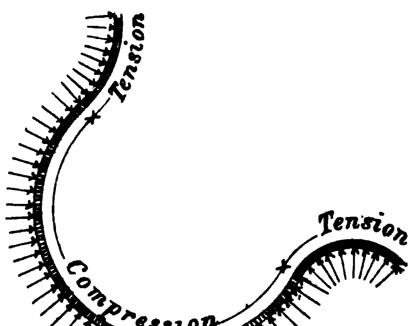


FIG. 179.

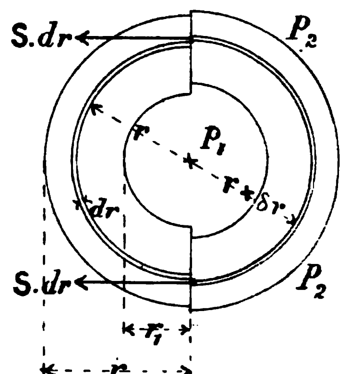


FIG. 180.

chamber backs; and the extra circumferential compression stresses which are thereby set up in the furnace saddles may have been the cause of occasional mishaps. The staying of these parts to the boiler shell will necessarily afford local relief, but very often leads to excessive bending strains in the screwed stay near the water level.

Thick Cylindrical Shells.—This subject would be out of place here were it not that some of the mathematical deductions will assist in estimating the distribution of stresses in riveted joints.

Let the left- and right-hand side of fig. 180 represent half the tube, respectively with and without the internal and external pressures p_1 and p_2. Then, dealing with the thin-walled ring dr of the radius r,

we find that if p is the radial pressure at this distance from the centre, and if S is the circumferential stress, we have

$S \cdot dr = (p + dp)(r + dr) - pr$. Neglecting $dp \cdot dr$, we have

$$(1) \qquad S - p = r \cdot \frac{dp}{dr}.$$

The radius of the ring has now decreased by

$$(2) \qquad \delta r = r \cdot \left(\frac{S}{E} - \frac{p}{\mu \cdot E}\right).$$

The thickness dr of the ring under consideration has decreased by

$$(3) \qquad \delta(dr) = + dr \left(\frac{p}{E} - \frac{S}{\mu \cdot E}\right).$$

If there are no longitudinal stresses in the cylinder, and if l is its length, then its alteration is

$$(4) \qquad \delta l = + \frac{l(S + p)}{\mu \cdot E}.$$

Differentiating (2) we have

$$\frac{d(\delta r)}{dr} = \frac{S}{E} - \frac{p}{\mu \cdot E} + \frac{r}{E}\left(\frac{dS}{dr} - \frac{dp}{\mu \cdot dr}\right).$$

But $d(\delta r) = \delta(dr)$, and combining this with (3)

$$(S - p)\left(1 + \frac{1}{\mu}\right) + r\left(\frac{dS}{dr} - \frac{dp}{dr}\right) = o.$$

Substituting the values of $S - p$ from (1) we find

$$(5) \qquad \frac{dp}{dr} + \frac{dS}{dr} = o.$$

It follows that $p + S$ is constant for all values of r, which shows that δl (4) is also constant, and need not be taken into account.

Differentiating (1) leads to the equation

$$(6) \qquad \frac{dS}{dr} - 2\frac{dp}{dr} - r\frac{d^2p}{dr^2} = o,$$

and combining this with (5) we get

$$(7) \qquad 3\frac{dp}{dr} + r\frac{d^2p}{dr^2} = o.$$

Integrating this and introducing the various constants leads to the following equations

$$(8) \qquad p + S = 2\frac{(p_2r_2^2 - p_1r_1^2)}{r_2^2 - r_1^2};$$

$$(9) \qquad p = \frac{(p_1 - p_2)\, r_1^2r_2^2}{r^2\,(r_2^2 - r_1^2)} + \frac{p_2r_2^2 - p_1r_1^2}{r_2^2 - r_1^2};$$

$$(10) \qquad S = -\frac{(p_1 - p_2)\, r_1^2r_2^2}{r^2\,(r_2^2 - r_1^2)} + \frac{p_2r_2^2 - p_1r_1^2}{r_2^2 - r_1^2}.$$

From this it will be seen that $p + S = o$ when $p_2r_2^2 = p_1r_1^2$. In

that case the metal is exposed to uniformly distributed spiral shearing stresses σ, as shown in fig. 181, and often seen around punched holes.

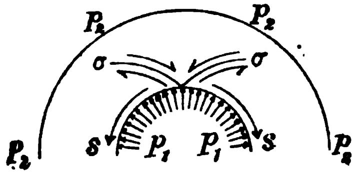


FIG. 181.

This condition is also approached, at least near the internal circumference, when r_2 is very large. For practical illustrations see 'C. E.,' 1893, vol. cxi. p. 212, and fig. 182, which represents a piece of a steel shell burst by a high explosive. All the fractured surfaces are sheared and inclined at an angle of 45°.

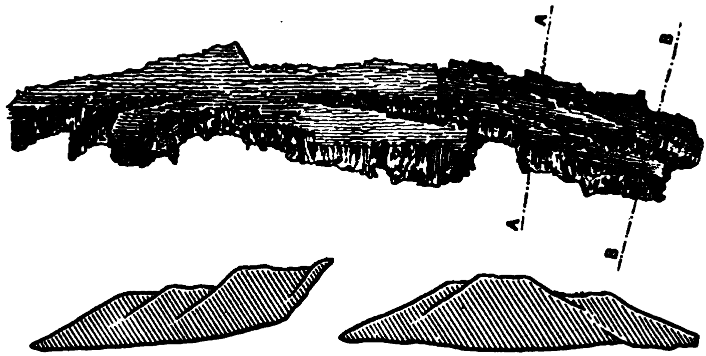


FIG. 182.

Other conclusions to be drawn are—

1st. The maximum stresses, either tension or compression, are always found at the internal circumference.

2nd. If the internal pressure $p_1 = o$, then the maximum circumferential compression stress is

$$S_1 = + \frac{p_2 \cdot 2 \cdot r_2^2}{r_2^2 - r_1^2}.$$

An approximately correct formula for cylindrical tubes would therefore be

$$S_1 = p_2 \frac{D + t}{2 \cdot t},$$

where D is the external diameter and t the thickness of plate.

3rd. If the external pressure $p_2 = o$, then the maximum circumferential tension stress is

$$S_1 = - p_1 \frac{r_2^2 + r_1^2}{r_2^2 - r_1^2}.$$

For boiler shell plates an approximately correct formula is therefore

$$S_1 = - p_1 \frac{D_1 + t}{2 \cdot t},$$

where D_1 is the internal and $D_1 + t$ the mean diameter of the shell.

In both these cases, therefore, the maximum stress has to be estimated as if the pressure were acting on an imaginary tube whose diameter is a little larger than the actual one.

These corrections are too unimportant to be taken into serious count in practice, except, as will shortly be explained, in the case of rivet holes.

The deformations of thick-walled tubes are found by inserting the values of S_1 and p_1 in equation (2) :

$$\frac{\delta r_1}{r_1} = -\frac{p_1}{E}\left\{\frac{r_2^2 + r_1^2}{r_2^2 - r_1^2} + \frac{1}{\mu}\right\}.$$

An approximate formula would be

$$\frac{\delta r_1}{r_1} = -\frac{p_1(r_1 + \frac{5}{6} t)}{E \quad t},$$

which shows that the dilatation of the internal diameter is almost proportional to the product of the external diameter into the internal pressure.

Plastic Tubes.—As regards the distribution of stresses in thick-walled cylinders when the plastic limit has been reached, little can be learnt until the plastic properties of steel subjected to compound stresses are better known. However, as the value of $\frac{1}{\mu}$ changes when this limit is passed from about $\frac{1}{3}$ to $\frac{1}{2}$, it is evident that it ought to be introduced into the various formulæ as a function of r, and then $\frac{\delta l}{l}$ is not any more a constant value, nor is $S + p$; and what complicates matters still more is that longitudinal stresses are set up which increase with increasing length of tube. The stresses in a gun barrel which is on the point of bursting are therefore not at all as simple as is usually assumed. A thorough analysis of the shape of the swelling of the material round a carefully drifted hole (fig. 183), similar to that carried out in the case of torsion tests, could perhaps be made to throw light on this subject.

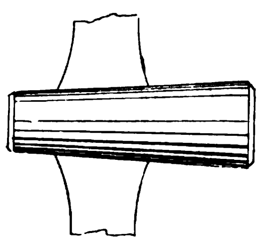


FIG. 183.

Riveted Joints.—At first sight it would appear that there is no simpler problem than to find the stresses in a riveted joint. Given the sectional area of the perforated plate, and the sectional areas of the rivets, the average stresses ought to be easily calculated. But these are only the average stresses. To find the positions and intensities of the maximum stresses is far more complicated, but worth examining, not only on account of the importance of the subject, but also because the difficulty of dealing with it more in detail will make it clear how little is yet known about other problems of boiler mechanics which, even at first sight, strike one as more complicated than this one.

Deformation of Rivets.—Fig. 184 represents part of a butt-strapped joint. A force Q is being transmitted from the central (shell) plate to the two butt straps, and in doing so the bearing pressures p_1 and p_2

come into existence. They tend to give the rivet a slight bend, and this deformation will cause an irregularity in their distribution, the pressure being proportional to the deformation. This is evidently a similar case to the one which presented itself when dealing with the influence of end plates on the distribution of stresses in cylindrical shells, and which, as was there shown, leads to the most complicated formulæ. Here it will be assumed that as long as the elastic limit has not been reached, p_2 is uniformly distributed over the thickness t of the shell plate, and p_1 is distributed over the thickness of the butt straps in the shape of a triangle. Then in the above case $p_1 = 2p_2 = \frac{2 . Q}{t . d}$.

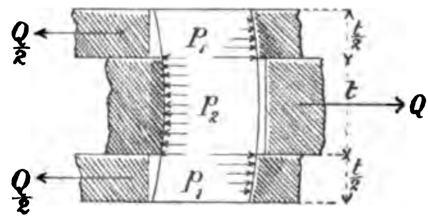


FIG. 184.

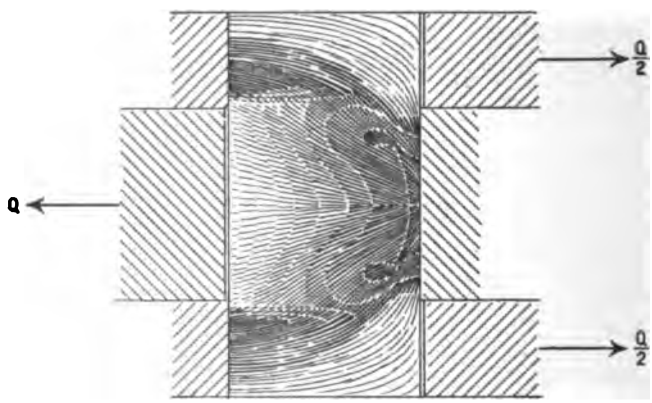


FIG. 185.

Stresses in Rivets.—These pressures produce bending and shearing stresses which, when reduced to right-angled resultants (see p. 132), act in directions which are indicated in figs. 184 and 185. In both the intensities are indicated by the shading of the various zones, and if placed over each other it would be found that for any particular point the two sets of stresses cross each other at right angles. Fig. 185 shows only compression stresses, and fig. 186 shows tension stresses at the left edge, which are gradually reduced and change into compression stresses at the right-hand centre. The stresses and their angles have been calculated on the assumption that the rivet diameter is equal to the thickness of the shell plate and twice as thick as the butt straps, and that the pressure is distributed as in fig. 184, but rounded off at the corners.

Then the mean stress along the line of shear, which is the mean of the tension and compression stresses, is 1·13 ton per ton of mean bearing

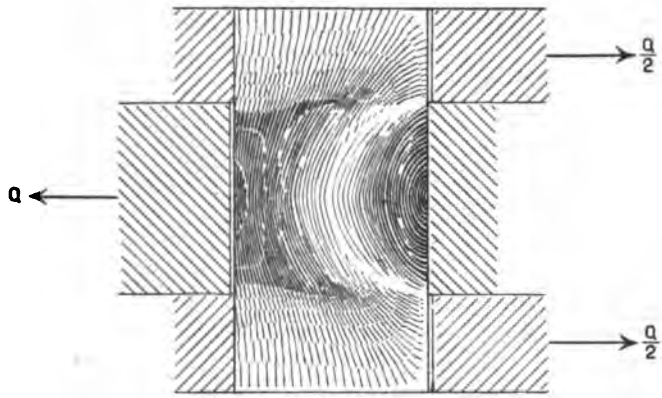


FIG. 186.

pressure, and the maximum stress in this line is 1·55 ton. On the upper part of the left edge there are some severe compression stresses, the maximum being 2·33 tons; and in the lower right-hand corner shrinking stresses are met with, consisting of two right-angled stresses of 1·2 and 1·45 ton per ton of mean bearing pressure. These diagrams and values might be used for the purpose of calculating the elastic deformation of the rivet, and the bearing pressures would then have to be modified; but, as no very important deductions will be drawn from these results, the more correct curves have not been determined.

Distribution of Bearing Pressure.—Another correction has to be introduced on account of the irregular distribution of the bearing pressure over the rivet diameter. Let the dotted line (fig. 187) represent the imaginary outline of the rivet section if it had been free to shift its position. But having come in contact with the circumference of the hole, which is shown by a black line, a pressure is called into existence, which may be taken to be proportional to the distance between the black and dotted lines. This normal pressure between the rivet and the plate is shown in fig. 188. $p = p_0 \cos \alpha$. Here p_0 is the maximum bearing pressure. The sum of $p \cos \alpha$ for the diameter d of the rivet is $\frac{\pi}{4} . p_0 . d . t$. This is equal to the load Q on the rivet, and the maximum bearing pressure at the centre line is

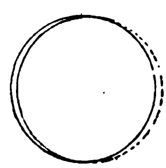


FIG. 187.

$$p_0 = \frac{4 . Q}{\pi . d . t} = 1{\cdot}275 \frac{Q}{d . t}.$$

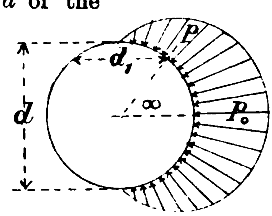


FIG. 188.

In the butt straps, as has been explained, this pressure will perhaps be double as much, depending on the flexure of the rivet and compression of the rivet hole.

Fig. 188 shows that $d_1 = d \cos \alpha$, and that if the rivet were divided

into numerous axial laminæ of equal thickness, the thrust on each one would be proportional to its own width, and if not connected amongst each other they would be deflected by different amounts (see figs. 189 and 190).

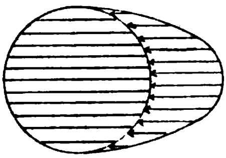


FIG. 189.

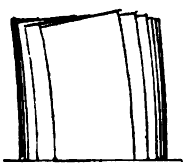


FIG. 190.

As these independent motions cannot take place there must exist a series of cross stresses in the rivets, which prevent the laminæ from slipping. It is also probable that in adjusting themselves to their surrounding conditions, the outer laminæ do not offer as much resistance as the inner ones, and therefore the normal pressure (fig. 188) is more likely to be distributed according to the formula $p = p_0 \cos a^3$, which, if true, would lead to the conclusion that the **maximum bearing pressure** to be found between the rivet and its hole is

$$p_0 = \frac{3 \cdot Q}{2 \cdot d \cdot t},$$

which is 50 % greater than the mean.

The Shearing Stress in a Rivet increases from nothing at either of its ends and at the centre of its length to a maximum at the plane of shear. In this plane it is greatest at the centre of the rivet, falling off in a parabolic curve towards the circumference. The maximum shearing stress in the centre is therefore about double the mean, viz.

$$\sigma = \frac{8}{\pi} \cdot \frac{Q}{d^2} = 2{\cdot}54\ \frac{Q}{d^2},$$

which coefficient is nearly the same as $1{\cdot}55 \times 1{\cdot}5$ as found above.

Shearing and Bending Stresses. If the leverage with which a load acts on a beam is equal to one-third of its thickness, then the limits of elasticity, both for tension compression and shear, are reached simultaneously (see p. 147). In a lap joint this condition most likely exists. In butt-strapped joints it will depend upon the thicknesses, and on the distribution of the bearing pressure, whether this is so or not. If the strap is equal to the thickness of the rivet, and the pressure distributed as in fig. 184, or if the thickness of the strap is $\frac{2}{3}$ of the rivet diameter, and p_0 is uniformly distributed over the length, then the above condition exists. In other cases the rivet gives way first, either by bending alone or by shearing alone. Of course these remarks only apply up to the elastic limit, and even here they are seriously modified by the various cross stresses which it has only been possible to hint at.

Plastic Rivets.—As soon as the stresses increase to such an extent that either at one point or another the elastic, or even the plastic, limit has been reached, then the conditions are altogether changed. All the pressures are more evenly distributed. The shearing stress is similarly affected, and is probably distributed as shown in fig. 165, p. 147. The ultimate strength of a rivet may, therefore, be estimated by those formulæ which give the mean stresses, while its working strength up to the elastic limit has to be found as explained above. It is, therefore,

very misleading to assume that the elastic limit and ultimate strength of a rivet stand in the same relation as the elastic limit and ultimate strength of a simple bar. In the latter case the ratio is about as one to two, in the former it is more nearly as one to five. A nominal factor of safety of five leads to a construction in which the limit of elasticity of a rivet is just reached with the ordinary working pressure.

When the dimensions of a rivet have been so arranged that the limit of elasticity of the metal is reached simultaneously at various parts, which is of course the most advantageous condition, then it does not at all follow that, on increasing the load, rupture will also take place simultaneously at all these points. In fact, there can be no doubt but that the bending moments increase at a relatively greater rate than the load, while the axial stresses due to them increase irregularly. While this is going on, the shearing stresses distribute themselves more uniformly and sink into relative insignificance, so that a rivet designed on the best principles for working conditions would rupture primarily through bending, and at a lower load than another which is so designed that all the ultimate stresses are reached simultaneously. Of course in this case the rivet would not be an equally efficient one under working conditions. That unexplained actions of this sort do exist is proved by nearly all experiments on this subject, for it will be found that it is not always the part which is apparently most strained that gives way. In fact, the stronger part—either the rivet or the plate—seems as if it were always endowed with extra strength. Thus a joint may tear through the plate when its stress is, say, 28 tons, while that in the rivets is 14 tons. But by reducing the sectional area of the latter it is possible to construct a joint in which the rivets will give way at, say, 20 tons, while the plate remains intact, though in this case it has been strained up to 32 tons or more.

Rivet Holes.—Circumferentially the bearing pressure in a rivet hole is of course distributed exactly in the same way as on the rivet. The stresses which are thereby produced in the surrounding material might be determined with considerable accuracy by a careful and exhaustive analysis; but this cannot be attempted here, and it will only be possible to recapitulate some of the views on this subject, and which are intended to be only approximately correct.

Firstly, the stresses between the rivet holes are supposed to be uniformly distributed. This is evidently incorrect.

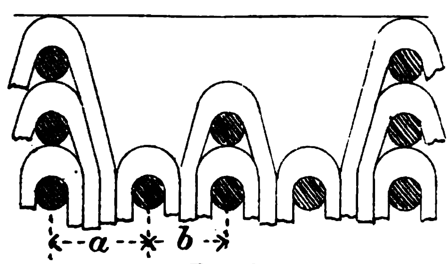


FIG. 191.

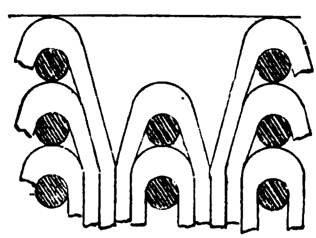


FIG. 192.

Secondly, the shell plates, or the straps, are supposed to be built up of a number of tapes or links, each one being of a sufficient section

for carrying the load which is placed on it (figs. 191, 192). The depths of these links over the rivets must be made 50 % greater than their widths. Although this view rather begs the question, it seems to give results which are well supported by practice, and if oftener applied would lead to modifications in some of the complicated riveted joints. Thus in fig. 191 it will be noticed that the inner row of rivets is irregularly pitched. This is necessary, as otherwise the tapes from the outer row would have to be split in two and spread out at an angle.

Thirdly, in order to estimate the amount of metal required between the rivet hole and the edge of the plate, this part is looked upon as part of a continuous beam.

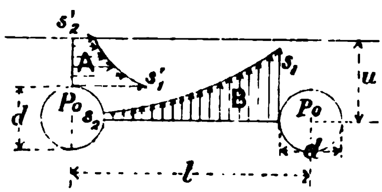


FIG. 193.

Fourthly, the circumferential shearing stresses are used as a basis.

Fifthly, a more thorough method is to deal with the metal surrounding the rivet as if it were part of a thick-walled cylinder.

Distribution of Stresses near Rivet Holes.—In fig. 193 the curves A and B represent the distribution of the circumferential stresses at these two points, due to the internal pressure p_o, which in this case is supposed to be uniformly distributed. In the formula (10, p. 159) for the curve A we have $r_1 = \frac{d}{2}$, $r_2 = u$, so that

$$S^1{}_1 = - p_o \frac{4u^2 + d^2}{4u^2 - d^2}.$$

And for the curve B we have $r_1 = \frac{d}{2}$, $r_2 = l - \frac{d}{2}$.

$$S_1 = - p_o \cdot \frac{4 . l^2 - 4 . l . d + 2 . d^2}{4 . l^2 - 4 . l . d}, \quad S_2 = - p_o \frac{2 . d^2}{4 . l^2 - 4 l . d}$$

Of course, as the adjoining rivet also produces stresses, S_2 has to be added to S_1, and we have

$$S_o = S_2 + S_1 = - p_o \cdot \frac{l^2 - l . d + d^2}{l^2 - l . d}.$$

The following tables contain some numerical values of the ratios of $S_1{}^1 : - p_o$, and of $S_o : - p_o$, for different values of d, u, and l :—

u : d	=	·75	1	1·25	1·5	1·75	2
$S_1{}^1$: $-p_o$	=	2·6	1·67	1·38	1·25	1·18	1·13

l : d	=	3	3·5	4	4·5	5	6	7	8
S_o : $-p_o$	=	1·17	1·11	1·08	1·06	1·05	1·03	1·02	1·02

From this it would appear that the tension stresses at the sides of

the rivet holes are slightly in excess of the rivet pressure p_o. Approximately

$$S_o = 2 \cdot \frac{Q}{d \cdot t}.$$

Similarly the stress $S^1{}_1$ may be expressed with sufficient accuracy by the formula

$$S^1{}_1 = \frac{3}{2} \cdot \frac{Q}{d \cdot t} \cdot \frac{1{\cdot}25}{\sqrt{\frac{u}{d} - \frac{1}{2}}}.$$

As has already been shown, these circumferential tension stresses combine with the radial pressures p_o to form shearing stresses, acting along spiral lines (fig. 182, p. 160); and, as mild steel gives way more readily under this stress than under either simple tension or simple compression, it is only natural that when the plate, and not the rivet, gives way, the line of fracture should start where shown in fig. 194; also compare fig. 181, p. 160.

It has been explained on p. 148 that, on account of the axial contraction of boiler shells, when subjected to internal pressure, the longitudinal seams are subjected to a compression stress. At the edges of the plates this will amount to about 30 % of the circumferential tension, and has to be subtracted from $S^1{}_1$, but it increases the spiral shearing stresses.

FIG. 194.

The above remarks apply only to the outer rows of rivets. In the second and third rows the conditions are far more complicated; but if, as is usual, the rivets are placed closer together, the stresses will be more uniform, and therefore relatively less severe.

Plasticity of the Solid Plate.—When the stresses have grown so intense that the plastic limit has been reached they will rearrange themselves; but instead of being distributed more uniformly, it is probable that they will become more local, the maximum stresses being found round the circumferences of the holes. Testing a set of joints to destruction cannot, therefore, give a correct idea as to the proportions which should exist between the thickness of the plates, the rivet diameters, and the holes.

Distribution of Stresses amongst several Rows of Rivets.—In fig. 195 the slanting position of the rivets is intended to indicate that a treble riveted lap-joint is being strained. If, as is generally supposed, each rivet bears the same load, then every one will be inclined to the same angle; but then the stretch, and therefore also the stress

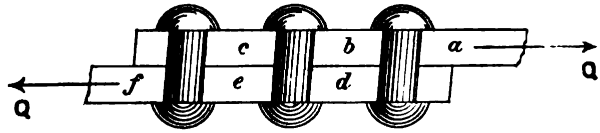


FIG. 195.

of the plate *b* would be equal to that of *d*, and the same for *c* and *e*. But it is obvious that, whereas the full stress is to be found at *a*, two-thirds will be found at *b* or at *e*, and only one-third at *c* and *d*, because in either case each rivet has only transmitted one-third of the load. These two distances have, therefore, only stretched half as much as *b* and *e*, and therefore the central rivet will not, as has been assumed, be inclined as much as the other two, and the shearing stress to which it is subjected will be less than the average, while for the two outer ones it will be more.

There are obvious reasons why it is easier to calculate the distribution of stresses in a butt joint than in a lap joint; but even this is a most laborious operation, and the results are not quite reliable. The following estimate was obtained with a joint whose dimensions were taken from practice, and which was intended to represent part of a double butt-strapped joint, having a percentage of 85·3 %. (See fig. 196.)

Assuming the circumferential stress on the shell plate to be 5·05 tons per inch, each rivet is supposed to be subjected to a mean stress of 5 tons per sq. in., and the mean stress in the plate between the rivets would be 5·92 tons. However, it was found that the mean shearing stress in the outer rivets is 5·10 tons, in the intermediate ones 4·20 tons, and 5·75 in the inner ones. This joint, therefore, is only equal to 74·1 % of the solid plate, which is a little less than might be obtained by using the far more practical double-riveted joint, having rivets 1¼ in. diam. and a pitch of 4·9 ins.

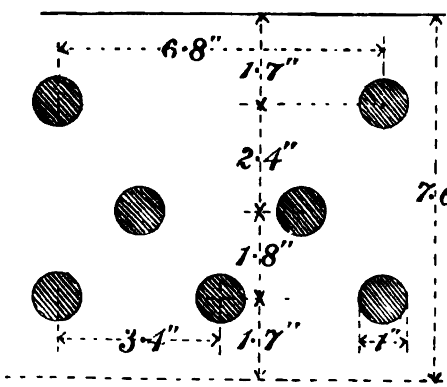


FIG. 196.

This should be a sufficient justification for the Admiralty practice, where no joint is credited with a greater strength than 75 % of the solid plate.

J. T. Milton ('N. A.,' 1885, vol. xxvi. p. 204) has recorded experiments on this subject which show that, even for the ultimate strength of a joint, considerable differences exist as regards the stresses in the various rows of rivets, and no doubt experiments on the elaborate joints to be found in bridges would show that their high percentage is only imaginary.

Experiments on Riveted Joints.—A very exhaustive list of experiments on riveted joints will be found, W. C. Unwin, 'M. E.,' 1881, p. 303. These and a few subsequent ones are contained in the following table :—

W. Fairbairn	1850 .	'Phil. Transactions,' vol. ii. p. 677.
E. Clark	,, .	'Britannia and Conway Bridges,' vol. i. ch. iv.

D. K. Clark	1858	'Recent Practice.'
J. Grantham	1860	'N. A.,' vol. i. p. 57.
D. Kirkaldy	1862	. . . . .
W. Fairbairn	1864	'Soc. Arts,' vol. xiii. p. 20.
N. Barnaby	1865–6	'Eng. Scot.,' vol. ix. p. 153.
J. Price	1869–70	„ vol. xiii. p. 47.
J. Cochrane	1870	'C. E.,' vol. xxx. p. 265.
W. Waller	1871	'N. Eng. I.,' vol. xx. p. 117.
W. R. Browne	1872	'M. E.,' p. 53.
J. G. Wright	„	„ pp. 77, 89.
Report	„	'N. Eng. I.,' vol. xxi. p. 67.
D. Kirkaldy	„	'Experiments on Riveted Joints.'
Sir W. Fairbairn	1873	'R. Soc. Edinburgh.'
B. B. Stoney	1875	'R. Ir. Ac.,' vol. xxv. p. 451.
J. Barba	„	Ch. iii.
J. Riley	1876	'N. A.,' vol. xvii. p. 135.
R. B. Longridge	1877	'Engr.,' vol. xliii. p. 125.
R. V. Knight	„	'C. E.,' vol. li. p. 131.
W. Boyd	1878	'M. E.,' p. 217.
B. Martell	„	'N. A.,' vol. xix. p. 12.
R. V. Knight	„	'C. E.,' vol. liv. p. 161.
Dr. H. Zimmermann	„	'Zeit. Bauk.,' vol. i. p. 530.
N. Barnaby	1879	'I. and S. I.,' p. 238.
D. Greig and M. Eyth	„	'M. E.,' p. 268.
R. V. J. Knight	1881	„ p. 720.
Prof. A. B. W. Kennedy	„	„ pp. 205, 232, 717.
„ „ „	1882	„ pp. 143, 242.
W. Parker	„	'C. E.,' vol. lxix. p. 50.
C. H. Moberley	„	„ vol. lxix. p. 337.
R. C. Longridge	1884	„ vol. lxxx. p. 154.
Prof. A. B. W. Kennedy	1885	'M. E.,' pp. 198, 249.
J. G. Wildish	„	'N. A.,' vol. xxvi. p. 179.
J. T. Milton	„	„ vol. xxvi. p. 204.
G. W. Manuel	1889	„ vol. xxx. p. 292.

The above contain chiefly those experiments in which actual joints were torn. Related subjects are sometimes to be found in these papers, but it will be more convenient to look for them under the following headings :—

Drilled and Punched Plates.

H. Sharp	1868	'N. A.,' vol. ix. p. 10.
J. Cochrane	1872	'M. E.,' p. 79.
J. Riley	1876	'N. A.,' vol. xvii. p. 135.
B. Walker	„	'M. E.,' p. 97.
A. C. Kirk	1877	'N. A.,' vol. xviii. p. 303.
B. Martell	1878	„ vol. xix. p. 1.
W. Parker	„	„ vol. xix. p. 172.
W. Boyd	„	'M. E.,' p. 222.
J. G. Muir	„	„ p. 285.
N. Barnaby	1879	'I. and S. I.,' p. 45.
„	1881	'M. E.,' p. 313.
E. Richards	1882	'I. and S. I.,' p. 43.
T. Wrightson	„	„ p. 49.
W. Parker	„	'C. E.,' vol. lxix. p. 50.
W. Beck-Gerhard	1884	'Gorni J.,' p. 347.
J. G. Wildish	1885	'N. A.,' vol. xxvi. p. 193.
P. D. Bennett	1886	'M. E.,' pp. 27, 44.
L. Tetmayer	„	'C. E.,' vol. lxxxv. p. 421.
— Rudeloff	1889	'Mitt. Berlin,' p. 97.

Shearing and Torsion.

D. Greig and Max Eyth	1879	'M. E.,' p. 268.
— Brock	1880	'N. A.,' vol. xxi. pp. 191, 204.
D. Greig and Max Eyth	1881	'M. E.,' p. 313.
E. Richards	1882	'I. and S. I.,' p. 11.
V. Appleby	1883	'C. E.,' vol. lxxiv. p. 268.
J. G. Wildish	1885	'N. A.,' vol. xxvi. p. 198.
Prof. W. A. B. Kennedy	,,	'M. E.,' p. 249.
J. Platt and R. F. Hayward	1887	'C. E.,' vol. xc. p. 382.
C. H. Carus-Wilson	1890	'Proceedings,' vol. xlvii. p. 363.

Friction of Joints.

D. Greig and Max Eyth	1879	'M. E.,' p. 268
Clark Kaven and Lavelley	1881	,, p. 327.

Diagonal Joints.

J. G. Wright	1872	'M. E.,' pp. 79, 90.
W. H. Shock	1874	P. 198.
'Enging.'	1887	'Numerous letters,' vols. xliii. xliv. 'Experiments,' vol. xliii. pp. 380, 428.

Theories about Riveted Joints.

J. H. Latham	1858	P. 1.
T. Baldwin	1866	'Soc. Eng.,' p. 150.
W. C. Unwin	1868	. . . . .
C. Reilly	1870	'C. E.,' vol. xxix. p. 454.
W. R. Browne	1872	'M. E.,' p. 53.
,, ,,	,,	'Engr.,' vol. xxxiv. p. 362.
Report	,,	'N. Eng. I.,' vol. xxi. p. 67.
W. R. Browne	1873	. . . . .
H. MacColl	1874–5	'Eng. Soc.,' vol. xviii. p. 111.
L. E. Fletcher	1876	'M. E.,' p. 64.
D. K. Clark	1878	Rules.
D. Adamson	,,	'I. and S. I.,' p. 392.
R. H. Twedell	1881	'M. E.,' p. 293.
G. Clauzel	,,	,, p. 167.
W. C. Unwin	,,	,, pp. 313, 333.
W. S. Hall	1885	,, p. 231.
J. A. Rowe	1884–5	'N. E. C. I.,' p. 73.
J. T. Milton	1885	'N. A.,' vol. xxvi. p. 204.

Factor of Safety.—Having dealt with the peculiar behaviour of materials under various conditions, and also with some problems connected with the mechanics of a boiler, it is necessary to add a few remarks on the term factor of safety, which was adopted at a time when practically nothing was known about the strength of materials, the distribution of stresses, and their influences; but even now the ultimate strength of a structure is taken as a standard, and it is assumed that if the loads which produced rupture were reduced to, say, one-fifth, then the stresses would be similarly reduced; but that this is not the case has been repeatedly explained in the last few pages. For instance:—

1st. In a tensile test piece the ultimate stress is a compound, resembling a negative fluid pressure, and this stands in no relation to the axial tension which exists in the sample at only one-fifth the ultimate load. (See fig. 132, p. 129.)

2nd. The working stresses in a narrow beam are relatively 33 % severer than the ultimate stresses. A nominal factor of safety of 5 is in this case actually only $3\frac{1}{3}$. (See p. 127.)

3rd. The working and ultimate torsional strength of mild steel stand in a similar relation, and in this case the nominal and actual factors of safety stand in the relation of 5 to $3\frac{3}{4}$. (See fig. 123, p. 126.)

4th. The shearing stresses in rivets under working conditions and during rupture have just been explained, and in this case the actual factor of safety is very much smaller than the nominal one. (See p. 165.)

As will be seen, a great deal depends upon whether the formulæ by which the stresses have been calculated are sufficiently comprehensive. If they are, and if the material is reliable, and the working conditions thoroughly well known, then there ought to be no reason for not reducing the factor of safety, for, from what has just been said, it is evident that some parts of a boiler are even now being worked with an actual factor of safety which is somewhat smaller than the real one, and there is no reason why other parts, where the nominal and actual factors are nearly equal, should not have the benefit, and might perhaps be made somewhat weaker than they are now.

It is also evident that when analysing the facts connected with the bursting, either experimental or accidental, of a boiler, it is very important to take into account the various deformations, for whereas elasticity of a structure is rightly looked upon as an extra guarantee of strength, plasticity throws the stresses into parts of a structure where they are least expected. To be thoroughly valuable, experiments on large structures should not only aim at obtaining information as to the weakest parts at the instant of rupture, but very careful measurements should be taken to determine the elastic deformations before that pressure is reached. (See p. 248.)

List of Boiler Tests and Explosions.

R. H. Thurston, 1872, 'Frankl. Inst.,' iii. vol. lxiii.

p. 89. Tested boiler to 82 lbs. cold, then burst at 90 lbs. hot.

p. 93. Stayed flat plates. Tested cold to 138 lbs., then burst at 165 lbs. hot.

p. 95. Tested box boiler to 60 lbs. cold. Vertical braces gave way; repaired these and tested hot. Reports were heard at 50 lbs. pressure; boiler burst at $53\frac{1}{2}$ lbs. pressure.

p. 99. Tested boiler to 200 lbs. at 100° F. Several braces broke under 115 lbs. steam pressure.

p. 99. U.S.A. steamer 'Algonquin' tested cold to 150 lbs. Some braces broke under 100 lbs. steam pressure.

R. H. Thurston, 1887, 'Explosions.' Tested boiler to 300 lbs. steam; burst at 235 lbs. on opening valve.

P. Carmichael, 'Eng. Scot,' 1869–70, vol. xiii.; also 1878–9, vol. xxii. Experimental bursting of a boiler.

L. E. Fletcher, 1876, 'M. E.,' p. 59. Experimental bursting of a boiler (cold).

W. Siemens, 'M. E.,' 1878.

D. Greig and Max Eyth, 'M. E.,' 1879. Experimental bursting of three boiler shells.

W. Parker, 'N. A.,' 1881, vol. xxii. p. 12. 'Livadia's' boiler burst under cold water test above working pressure.

Ibid., 'N. A.,' 1885, vol. xxvi. p. 253. Shell boiler burst under hydraulic test.

Ibid., 'N. A.,' 1889, vol. xxx. p. 297. Experimental bursting of a boiler (cold).

J. Scott, 'N. A.,' 1889, vol. xxx. p. 285. Experimental bursting of a Navy boiler.

Parliamentary Committee on Boiler Explosions, 1817.

Report on boiler explosions. Rep. Comm. Parl., 1849.

Parliamentary Committee on Boiler Explosions, 1871. Parliamentary Reports No. 186, vol. lxvi. p. 43, No. 378, vol. lxvi. p. 85; 1877, No. 361, vol. lxviii. p. 373.

Martens, 'Rep. S.U.A.,' various dates.

'Board of Trade Reports on Boiler Explosions.' Amongst these latter the following are of interest:—

1861, part iv. Locomotive boiler tested to 196 lbs. at 162° F.; burst seven months later under 120 lbs.

'Acrefair,' Dec. 10, 1880. Boiler locally weakened to 61 lbs. permissible working pressure; burst at 32 lbs.

No. 228. Drying cylinder burst. Factor of safety about 20·3.

No. 229. Lancashire boiler shell burst. Factor of safety was about 2·8.

No. 237. Locomotive boiler burst.

No. 243. Boiler shell burst through manhole at 100 lbs., after having been recently tested to 150 lbs. steam.

No. 249. Boiler shell exploded. Factor of safety 2·75.

No. 252. Boiler shell exploded. Factor of safety 11.

No. 265. Boiler shell exploded at 83 lbs. It had recently been tested cold to 92 lbs.

No. 314. Boiler shell exploded. Factor of safety 3·5.

No. 346. Furnace collapsed at 50 lbs. It had been tested to 95 lbs. only three days previously.

Other interesting cases are—Explosion on the steamer 'Mülheim No. 5,' 1866; explosion on the steamer 'Parana,' 1869; explosion on the steamer 'America,' 1871; explosion of a locomotive boiler, 'Engineering,' 1890, vol. l. p. 332.

T. W. Trail ('C. E.,' 1877, vol. li. p. 31) mentions the bursting of two boilers whose factors of safety were 3·9.

L. E. Fletcher ('C. E.,' 1884, vol. lxxx. pp. 136 and 139) mentions several curious explosions.

J. J. Platt, 'M. E.,' 1878, p. 260. A steel fire-box plate cracked after being in use five months.

McFarlane Gray ('N. A.,' 1877, vol. xviii. p. 326) mentions that a boiler intended for a working pressure of 30 lbs. was tested to 400, repaired, and then worked all right.

J. A. Rowe, 1884, p. 7. A boiler burst at 30 lbs. working pressure after a cold water test to 59 lbs.

CHAPTER VII.

BOILER CONSTRUCTION.

In the following pages the various workshop practices of boiler construction will be dealt with as briefly as the subject will allow. The question of cost cannot, of course, be entered upon, but occasional reference will be made to the time required for the various operations; this necessarily varies in different works, depending not only on the perfection of the machinery, but also on the skill and energy of the men. More attention has been paid to the different practices and tools for obtaining the same object, and occasionally methods have been mentioned which are practically obsolete or not in use, but which may have done good service or must be looked upon as warning examples. Naturally every modern device has not been discussed, but it is hoped that none except unimportant ones have been neglected.

The order in which boiler construction will be taken is to deal firstly with the various operations to be performed on the boiler shell, then with the internal parts, and then with the boiler as a whole.

The plates as they arrive from the rolling mills are never exactly of the specified sizes, being usually about $\frac{1}{2}$ in. larger in each dimension. The thickness is kept within reasonable limits by the condition that any excess weight beyond, say, 5 % margin will not be paid for; but difficulties are sometimes occasioned by stipulations that certain weights shall not be exceeded which, when compared with the plate thicknesses, are too low. The weight of 1 sq. foot of iron is equal to about 40 lbs. per inch of thickness, and 41 lbs. for steel.

The edges of plates are nearly always thinner than the centres. This is almost unavoidable, for as soon as the rolls are set so that the conditions are reversed the plates are seriously puckered. It is not possible to give a full explanation without entering into unnecessary details, but there is no difficulty in understanding that if during the last pass through the rolls the thickness of a plate is reduced $\frac{1}{32}$ in., while its thickness is $\frac{1}{2}$ in. at the edges and $\frac{7}{16}$ at the centre, then these parts will stretch relatively $\frac{1}{16}$ and $\frac{1}{14}$ of their length, and the centre must pucker. Should the rolls have been made quite parallel, then as soon as they get heated their centres swell and produce the above result. If they are made slightly hollow, the plates which are rolled first will be thin at the edges, and perhaps frilled; but the greatest trouble is experienced when the widths of the plates to be rolled vary much, for an enormous pressure (about 500 to 1,000 tons) has to be exerted, and the spring in the rolls is very appreciable.

Shearing Operations.—In a boiler shop large shearing machines are not required, but certain plates, such as those for the boiler ends and for the combustion chambers, are often ordered with a comparatively large margin, on account of flanging, and this excess is then most easily removed by shearing. A small but powerful machine, capable of cutting steel plates up to $1\frac{1}{4}$ in. thick, will be found a very handy tool. The levers and handles for working it should not project in front; otherwise they may come in contact with such of the plates whose flanges have to be sheared while standing on end. The arrangements for replacing the shearing blades should be such that this can be done quickly, so that, as occasion arises, curved or cornered shears may be substituted for straight ones.

The question as to how much material ought to be planed off sheared edges in order to remove the injurious effect is, and probably will remain, an unsettled one. Some engineers look on shearing and punching as being perfectly harmless; others insist on subsequently planing away at least $\frac{1}{4}$ in. Probably the experiences which led to these diverging views are due to the use of various qualities of material. Remarks on this subject will be found in the chapter on 'Strength of Materials,' and the conclusions arrived at there are that good material is not injured, while bad material grows brittle, and the more so the thicker the plates are. Evidence will also be found there in support of the view that the brittleness caused by shearing gradually extends into the plates. Under any circumstances it is well to guard against possible failings of this sort by insisting on a cold bending test of samples with sheared edges, or of samples with punched holes. In this latter case a standard punch and bolster should be determined upon, as their relative diameters influence the results. If a sample which has been punched or sheared bends well when cold, and particularly if it does so after having been put aside for a week or more, then there will be little fear that the material can satisfactorily withstand this and the less severe workshop treatment.

Punching Operations.—The effects of punching are so very similar to those produced by shearing that nothing need here be said about

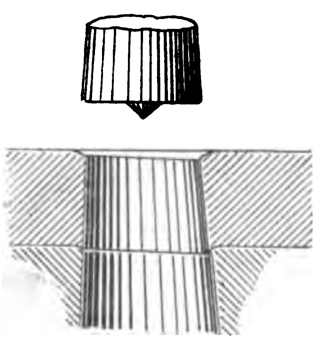


FIG. 197.

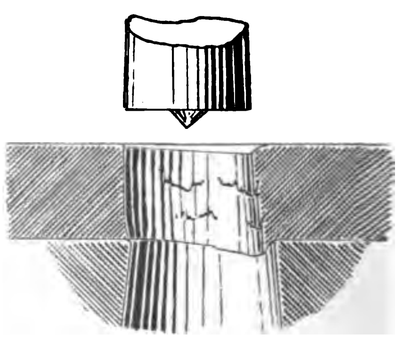


FIG. 198.

them, except, perhaps, that it would be better if no punching machines were used in a boiler shop, so that all holes could only be drilled.

However that is rarely, if ever, the case, and therefore it is as well to fit as strong and accurate a machine as can be obtained, and to insist on carefulness in setting the bolster. If carelessly placed the punched hole will be a very irregular one (see fig. 197). This is particularly the case if the guides of the punching press slide are a loose fit, for then it will happen that one hole is quite fair, while the next has the above form, or even a worse one (as in fig. 198), due to the punch having started penetrating while its edge was actually overlapping the circumference of the die. The breaking of punches under such conditions is not to be wondered at.

The clearance, or rather the difference, between the diameters of the punch and of the die hole is usually about 10 to 15 %. This agrees fairly well with the following published recommendations:—

W. H. Shock (1880, p. 166) gives 15 to 20 %. J. H. Wicksteed ('M. E.,' 1878, p. 244) recommends one-sixth of the thickness of plate. This amounts to 11 % where the diameters of the holes are half as large again as the thickness of the plate, and 17 % when they are equal.

Spiral and Slanting Punches have been used not only to lessen the injury to the plate, but also to reduce the pressure on the machine. W. Barr (1880, p. 92) mentions that Kennedy's spiral punch (fig. 199) requires only one-third the power of an ordinary one. Stern ('M. E.,' 1878, p. 239) gives the following information:—

—	Diam. of Hole	Punching Force	Tenacity of Punched Plate
Ordinary punch . . .	$\frac{7}{8}$ in	33 to 35 tons	26 tons
Spiral „ . . .	$\frac{7}{8}$ „	22 „ 25 „	$28\frac{1}{2}$ „

L. Hill ('M. E.,' 1878, p. 244) mentions slanting punches (figs. 200, 201).

It is only natural that, in comparison with ordinary punches, less force, though possibly just as much power, is required with spiral or

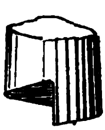


FIG. 199.

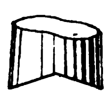


FIG. 200.

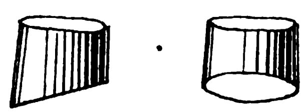


FIG. 201.

slanting punches, because with them only a small part of the circumference is doing work, just in the same way as less force is required in a shearing machine if the cutting blade is set at a steep angle. When large holes have to be punched, there ought to be no hesitation in substituting one of the above punches for the ordinary flat one. Fig. 200 seems to be the best suited for hand holes, &c., not only because of its being symmetrical, but because the greater part of the edge is shaped like a good cutting tool. The reverse or V shape would in this respect be the worst possible. The spiral punch leaves a mark in the circumference, which in large holes may be an inconvenience.

A favourite plan in some works is to punch the holes in all those plates which for some reason, such as flanging, have to be annealed, but,

on account of the blindness of some of these holes when fitted together, a great deal of hand labour has to be expended on them in the way of chipping and rimering, so that if added together the expenses for such holes would be found greater than if they had been drilled; besides they will not be satisfactory jobs.

The holes of the seams in the furnaces are also sometimes punched; but here, particularly with the circumferential seams at the front ends, there is danger that the drawing of the furnace mouth, to meet the flanged front plate, will produce cracks.

Various seams in the flat plates of the front and back ends are sometimes punched, even when they are $\frac{7}{8}$ inch thick; but the warping of the plates and the certainty of having to chip and rimer a large number of holes ought to be a sufficient objection to this practice.

The lower edges of the back tube plates are almost invariably punched, leaving them quite ragged. In some works they are left in this condition, as it is useless to caulk this edge. For appearance sake most boiler-makers chip it. Under any circumstances it would be best to leave a good bevel, so as to prevent steam lodging there and causing the saddle seam to heat. By using a square or rectangular punch such edges would be left in a better condition for chipping.

Punching is also resorted to for producing holes for guiding the trepanning tools with which the tube plates are bored. This is a bad practice, and leads to irregularities amounting to $\frac{1}{8}$ in. in the various diameters of the finished holes, and even affects their roundness.

A less objectionable though still unsatisfactory use of the punch is the practice of perforating those points of the back end plates and of the combustion chamber plates where the screwed stays are to be fitted. When placed in position drills and then taps are passed through them, which remove all brittleness, even if the plates have not been annealed after punching. (For injury done by punching see p. 169.)

Illustrations of various types of shearing and punching machines will be found in the following publications:—

'Engineering,' vol. xxxix. p. 219, 'Hydraulic Shearing Machine'; vol. xlii. p. 221, 'Portable Pneumatic Punching Machine'; vol. xliv. p. 16, 'Shearing Machine'; vol. l. p. 688, 'Punching and Shearing Machine'; vol. l. pp. 177–179, 243, 247, 494, 519, 'Punching Machines.'

Planing Operations.—Illustrations of plate-edge planing machines will be found in 'Engineering,' vol. xxxvi. p. 384; vol. xlix. pp. 245, 252; vol. l. pp. 536, 625. All the shell plates have to be planed at their edges. They are bolted down on the planing machine (fig. 202), and a tool in the slide rest R_1 travels along its edge, at a speed of about 10 to 18 ft. per minute, mean speed 12 ft. with a feed of $\frac{1}{32}$ in. for 1-in. plates, and proportionately more or less for thicker or thinner plates, removing the superfluous material at the rate of 5 to 6 cubic inches per minute. To this has to be added the time for setting the plate. The total time required to set and plane four shell plates measuring 20 ft. $\times 5\frac{1}{2}$ ft. $\times 1\frac{1}{8}$ in., equal to 102 ft. running, amounted to about 12 hours, from which it is clear that the proper setting of the plates, which had to be repeated eight times, takes up very much more time than that required for the removal of the material. The machine

used had two sets of frames, placed at right angles (see fig. 202). With the old-fashioned machines, in which the plate has to be reset for each

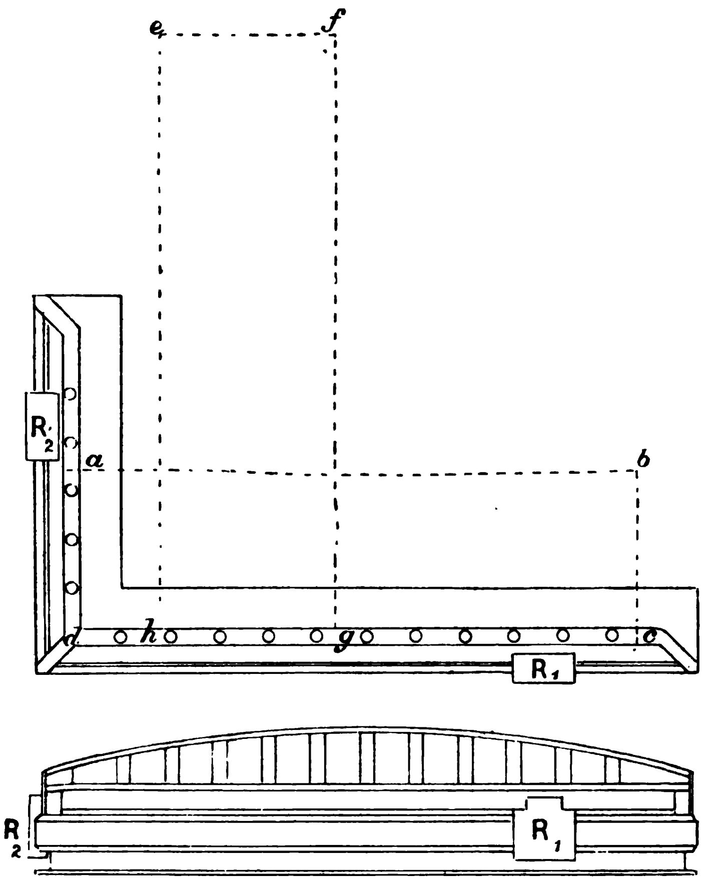


FIG. 202.

edge, the time required would have been twice as long, and much more space would have been required, for the plate, which is shown by the dotted lines, *a*, *b*, *c*, *d*, would have to be turned round to the position *e*, *f*, *g*, *h*.

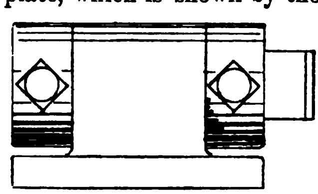


FIG. 203.

A little saving in time is effected if the planing tool be so fitted that it can be turned round, and cut both during the forward and backward travel (see fig. 203). But when the edges have to be bevelled the tool must have two cutting faces (see fig. 204).

The amount of bevel of edges to be caulked varies from nothing to

1 : 3 (see p. 240), but for the butts of shell plates it should be about 1 : 30 ; otherwise they will not close properly (see fig. 205).

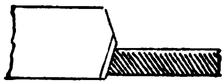


Fig. 204.

Should the planing machine be shorter than the plates, their edges will have to be planed in two operations. The usual plan is to withdraw the tool gradually when it reaches the end of the stroke, but in some works part of the material at this point is first removed by chipping.

In old machines, where the holding-down frame is secured at its

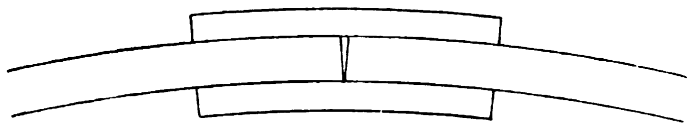


Fig. 205.

ends, as shown in fig. 206, only definite lengths of plates can be planed, unless a bracket is bolted to the bed.

Large plates are generally ordered with a margin of $\frac{1}{4}$ inch all round, and this is occasionally exceeded by another $\frac{1}{4}$ inch when shearing. All this material has to be removed on the planing machine, and great care has to be taken that the final dimensions are correct. They should never be marked off with anything else but a steel or iron rule, for the differences of expansion of wood, iron, and brass are very appreciable on large dimensions. The working drawings should always contain the widths and lengths of the plates, though the latter dimension is generally omitted. To find the exact circumferential length of a shell plate multiply the *mean* diameter of the particular strake by 3·1416.[1]

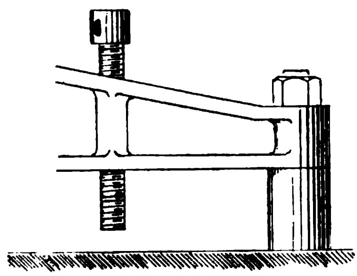


Fig. 206.

This length has to be divided by two, three, or four, according to the number of joints in the shell. The length of the adjoining strake may be found in the same way, but should be checked by adding or subtracting 6·283 times the mean thickness of the two plates as measured at their overlapping edges. It is necessary to be accurate on this point, otherwise the plates will not butt properly. In lap-jointed boilers one width of the lap has to be added to the length of each plate.

The furnace and combustion-chamber side plates are also planed on these machines, and their dimensions are either marked on the drawings or have to be measured from them. The lengths should be found by measuring the circumferences of the flanged plates and furnaces when fitted together.

The flat edges of the front and back end plates are also planed in these machines, but only after having been flanged and fitted together.

Drilling Operations.—In all first-class works every hole in a boiler

[1] A rough approximation to this value is $\frac{22}{7}$ where very great accuracy is desired the fraction $\frac{355}{113}$ may be used.

is drilled, and generally drilled in place. A considerable amount of ingenuity has therefore been expended in designing machines which will do this work with speed and accuracy, and which can be adapted to various uses. Naturally a great many different patterns are in use, as will be seen from the following list :—

W. S. Hall, 'M. E.,' 1878, p. 565. Drilling machinery. He mentions Hutchinson's, Welch's, Buckton and Wicksteed's, and Buckton's multiple drilling machines ; Adamson's, Dickinson's, Jordan's, and Kennedy's drilling machines ; Hall's portable, Brown's and Thorn's steam drilling machines ; McKay's equilibrium drill and other furnace and manhole boring tools, and also Shaw's flexible shaft.

W. Arrol ('M. E.,' 1887, p. 312) describes the drilling machinery used in the construction of the Forth Bridge. This paper is well worth studying, though not intended for boiler work. In 'Engineering' will be found sketches of the following.

Ordinary drilling machines, vol. xxxiii. p. 348 ; vol. xxxv. p. 99 ; vol. xlvi. p. 327 ; vol. xlviii. p. 191.

Radial drilling machines, vol. xxxiii. p. 134 ; vol. xxxviii. p. 388 ; vol. xxxix. pp. 57, 360 ; vol. xl. p. 246 ; vol. xli. p. 29 ; vol. xliii. p. 269 ; vol. xliv. p. 42 ; vol. xlvi. p. 468 ; vol. xlviii. p. 180 ; vol. xlix. p. 248 ; vol. l. p. 184.

Multiple drilling machines, vol. xxxiv. p. 373 ; vol. xliii. p. 69 ; vol. xliv. pp. 150, 289, 292 ; vol. xlv. p. 451 ; vol. xlvi. p. 90 ; vol. xlix. pp. 250, 251.

Shell and back end drilling machines, vol. xxxiii. p. 586 ; vol. xl. p. 424 ; vol. xli. p. 620 ; vol. xlii. p. 420 ; vol. xlviii. p. 501.

Portable drilling machines (for furnaces), vol. xxxii. p. 162 ; vol. xxxix. p. 570 ; vol. xlii. p. 637 ; (for shells) vol. xl. p. 320 ; (hydraulic) vol. xliii. pp. 130, 131 ; vol. xliv. p. 295 ; vol. xlvii. p. 644 ; (hand) vol. xlv. p. 185. Ring-shaped shell drilling machine (Forth Bridge), vol. xxxix. p. 57 ; vol. xlix. p. 248.

Combined multiple and radial drill, vol. xlii. p. 613.

Drilling and tapping machine, vol. xlviii. p. 501 ; vol. l. p. 184.

Drill grinders, vol. xl. p. 320 ; vol. xliii. p. 101 ; vol. xliv. p. 6 ; vol. l. p. 674.

These numerous references make it needless to discuss the distinguishing features of the various types, and attention will only be drawn to a few points.

The Ordinary Drilling Machine is used for little else but small or occasional jobs. Its table can be raised or lowered, and can generally be turned round a vertical axis. The feeding is done by hand. In shipyards, where these machines are used for countersinking holes, the feeding arrangement is a balance lever.

Radial Drilling Machines are made in various forms. Sometimes they are fixed against walls, and are driven by belting from above, or they stand alone and are driven from underground. In that case they are generally made so that their arm can sweep through 360°. In some cases the table is adjustable, in others the arm can be raised or lowered. Arrangements for doing this by power are a great convenience, and all moving parts should be balanced. It is well to make the balance weights extra heavy, so as to reduce the slack of the drill to a minimum.

Multiple Drilling Machines are passing out of favour in boiler shops, because their chief occupation, drilling shell plates before bending, is gone. They are generally arranged to slide along a frame, and each spindle can be shifted and worked independently of the others, or they can be set to the proper pitch and moved along together. In some machines the spindles can also be moved at right angles to the frame, so that a double row of holes can be drilled without resetting the plate. In these machines the various spindles cannot of course be brought close together, which causes a slight waste of time at the two ends of a seam, but is otherwise an advantage. In another type of machines the spindles are placed very close together. Figs. 207, 208 show two arrangements for adjusting the pitch in these cases. A turn of the screw S (fig. 207) will separate the nuts N_1, N_2, N_4, N_5 from the centre

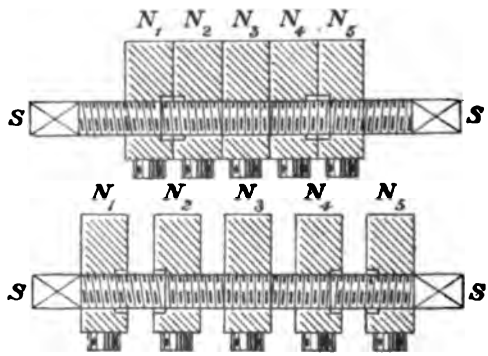


FIG. 207.

one N_3, and they are then clamped. Or a turn of the spindle C (fig. 208) will turn the hollow right- and left-handed screws z_1, z_2, z_3,

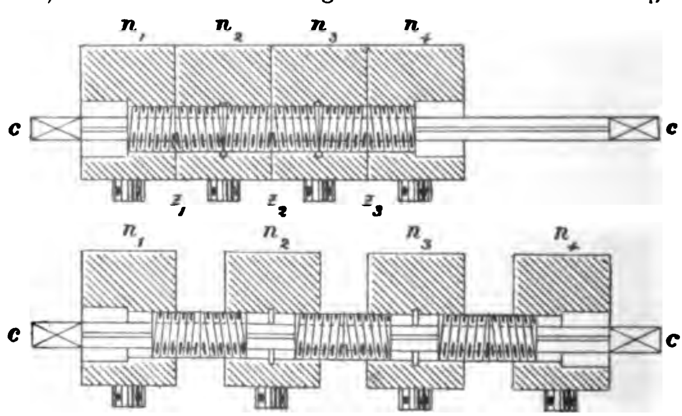


FIG. 208.

which connect the four nuts n_1, n_2, n_3, n_4, and these are then clamped The drill spindles are attached to these nuts and move with them

Should one of the drills break, or be put out of use, all the others would of course have to be stopped for a time.

Shell Plate Drilling machines usually consist of two vertical columns, of which one or both are movable. One, two, or even three drill spindles are attached to a slide, which can be moved up or down these columns into any required position. If shells are to be drilled they are placed on a turntable between the two columns (sometimes there are three or four). Much time is wasted over the longitudinal seams, for usually only one can be attacked at a time, and in this respect it is a great advantage if each column can be moved both in a radial and in a circumferential direction, even if only through a limited range. Then the angular adjustment need not be done by the turntable.

Another shell drilling machine is shown in fig. 209. The boiler shell rests on four wheels, W, W. A slide rest S is movable along the bed plate, and carries an arm A, which can be set to any angle by means

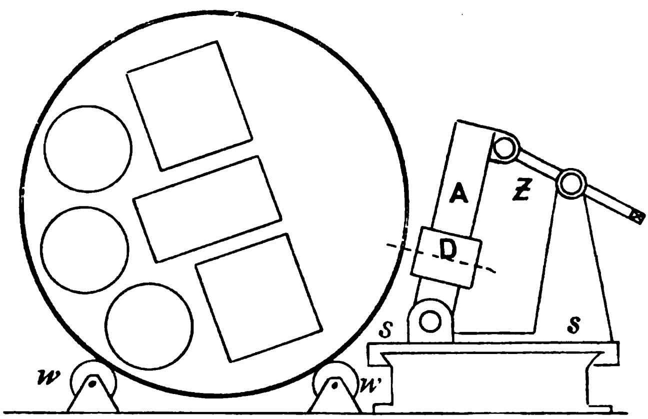


FIG. 209.

of the screw *z*. It carries the drilling spindle D, which can be moved up or down. It is clear that the drills—for there are usually two—can, within certain limits, be set to any required angle or position. When all the holes in a given span are drilled, the boiler is turned round the necessary angle and a fresh start made.

Boiler Back End Drilling machines usually consist of two columns, both movable along a strong bed plate; the drill spindles can, as in one of the previous cases, be raised or lowered to any convenient height, so that every part of the back end plate of a boiler could be drilled if placed in the proper position. These machines are sometimes fitted with the necessary gear for tapping holes and screwing in the screw stays, and in some cases circular saws can be attached, with which to cut off the projecting ends.

A very convenient machine for drilling furnaces in place is shown (fig. 210). A frame with three or four set screws is firmly screwed into the mouth of the furnace; a light drilling machine is attached to the centre of this frame in such a manner that it can be clamped in any radial position. The driving wheel projects beyond the mouth of the

furnace, and is driven by a belt or gut rope. A somewhat similar machine, but fixed, is sometimes used for drilling the furnace

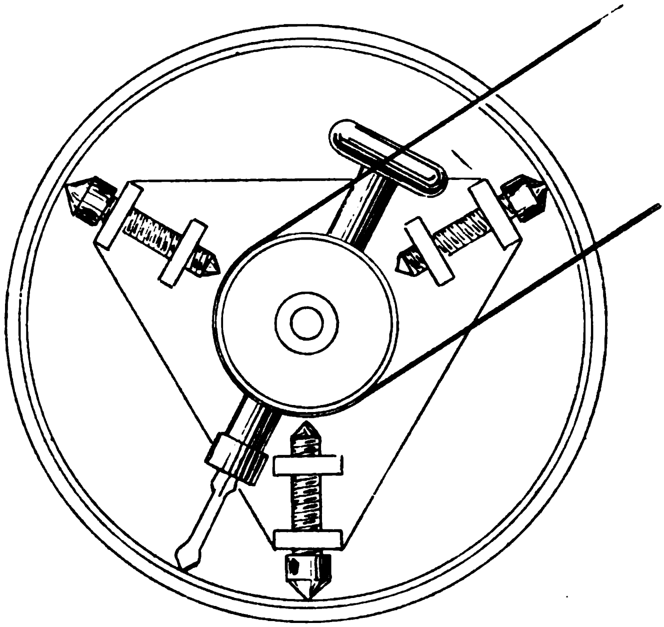


FIG. 210.

fronts out of place after the furnaces have been fitted and the holes marked off.

Any one of the more powerful drilling machines can be used for boring the tube plate holes. A small hole is first drilled or punched in the centre. The guiding spindle S (fig. 211) of a special tool holder, which is secured to the drilling machine, is inserted into one of them, and the cutting of the circumference of the large hole is then done by the parting tool T while it revolves. P is the tube plate which is to be bored. Unless the spindle S is a very good fit in its guide hole, the tube holes are very apt to be irregular both as regards shape and size, and as the setting of the tool T is a somewhat tedious job, it is not to be wondered at that every time it has to be done the diameter of the hole changes. This can easily be demonstrated by measuring several holes, which will be found to differ sometimes by as much as $\frac{1}{16}$ in. even in the same plate. No wonder, then, that tubes which are somewhat less in diameter than the smallest holes should split while being expanded into the larger ones. In some machines S is replaced by a sliding centre which rests in a large centre punch mark and guides the cutter.

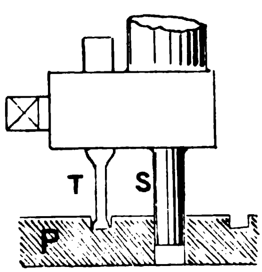


FIG. 211.

Other contrivances related to drilling machines are furnace and manhole boring tools. If intended only for the former of these purposes, they are practically nothing but very powerful drilling machines ; only, instead of a drill, an arm with a slide rest is attached to the vertical spindle. A parting tool is secured to this rest, as shown in fig. 212, and the furnace front plate bolted in the desired position on the bed plate and bored out. Most of these machines are now arranged in such a manner that the vertical spindle can be made to travel horizontally backwards and forwards, whereby the hole cut into the plate will be an elliptic one, as required for manholes.

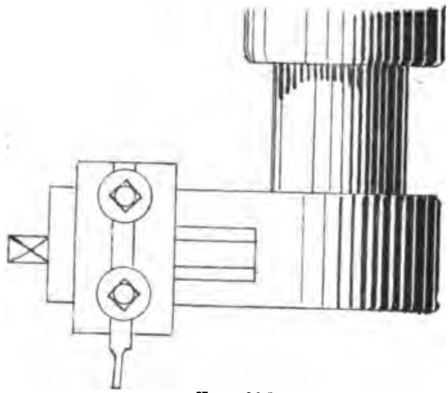


FIG. 212.

As regards the drills themselves, something may be learnt from W. S. Hall's paper ('M. E.,' 1878, p. 565), as well as from W. F. Smith's (ibid. 1883, p. 56), who discusses the cutting angles of tools and drills and speeds.

Time required for Drilling.—To drill a hole 1 in. deep per minute, and to spend about half a minute for setting, seems to be a fair allowance for holes of from $\frac{3}{4}$ in. to $1\frac{1}{4}$ in. diameter. W. F. Smith gives a circumferential drill speed of 20 ft. per minute with $\frac{1}{100}$ in. feed, while J. H. Wicksteed, in the discussion which followed, recommends 40 ft. and $\frac{1}{200}$ in., so that as regards speed of feed the result is the same in both cases. For 1-in. holes this means a drilling speed of about $1\frac{1}{2}$ in. per minute, which is a very high value.

Hand drilling is much slower, being at the rate of about 5 to 6 ins. depth of 1-in. holes per hour, and about 3 minutes for setting. Smaller holes can be drilled more quickly, and, generally speaking, the labour is proportional to the weight of metal removed. The limiting conditions for automatic feed are the strength of the drill and the clearance for the borings. In both respects the twist drill is superior to the ordinary one, but very few boiler shops have retained it in use. It is stated that after a time these drills wear away near their ends, and grow taper, and get jammed in the holes and break. Another and probably more powerful reason is that sufficient care is not taken to run them perfectly true, nor to guide them as required, nor to sharpen them correctly ; and if all or nearly all the work is thrown on one cutting edge, it cannot be expected that the result will be a satisfactory one. The irregular action which takes place with ordinary drills, if one cutting edge is left longer than the other, is sometimes made use of to produce taper holes. The deeper the drill penetrates when in this condition, the larger grows the diameter of the hole. This is of course impossible with twist drills, as they are guided by the hole they make.

Irregularly-shaped Holes.—Even ordinary drills are very liable

to produce irregularly-shaped holes. The action which takes place is easily understood by examining fig. 213, in which a two-cornered spindle moves freely, but without slack, in a three-cornered hole. A three-cornered spindle would do the same in a four-cornered hole (fig. 214), and this principle has been made use of for producing square holes.

As regards the general practice, it will be found that in nearly all works the shell plates are drilled after they have been bolted together with the flanged end plates fitted into them, the holes being drilled through shell plates and flanges in one operation. A few years ago the circumferential shell seams were often drilled before the plates were bent, and considerable ingenuity was displayed in devising means for ensuring that the holes in the two strakes should correspond. But this practice has died out, except in small works, and vertical multiple drilling machines are now used chiefly for the holes in the seams of the end plates and of the combustion chambers. If, as is usual, several plates are being drilled at a time, they should be screwed together by properly fitted bolts, and not clamped only; otherwise they slide on each other, and the holes of the lower plates will be out of place. Twist drills should be used for this work, because ordinary ones do not drill in a straight line.

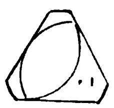


FIG. 213.

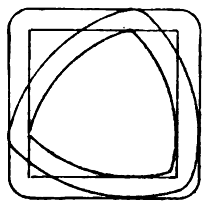


FIG. 214.

Marking off Holes.—Should it be necessary to drill shell plates before bending, it is very important to make accurate templets both for the inside and outside strakes, comprising about thirty or forty holes, and the exact positions of several holes should be carefully calculated as a check on the templets. In works where this plan is adopted the necessary appliances for drilling in place are doubtless wanting, and then the holes in the end-plate flanges and in the longitudinal seams will have to be drilled by hand, the outer machine-drilled holes acting as guides. The latter should never be drilled before bending the plates, as they are thereby seriously weakened at points where strength is of the utmost importance.

Great care should always be taken that these holes cover each other perfectly, for if blind, the sectional area of the rivet in the plane of shear is reduced, or if this is put right by chipping and rimering, the section of the plate round the hole is reduced.

Another practice is to punch or even to drill the holes of a smaller diameter than required, and to drill away the superfluous material when fitted together.

Bending Operations.—Since the failure of the steel shell plates of the steam yacht 'Livadia' there exists a very justifiable dread of bending such plates while hot, but as long as this operation is not carried out at a blue heat the plates ought to suffer no permanent injury, and where the rolls are not sufficiently strong to bend cold plates they will have to be heated. It must not be forgotten that the 'Livadia' case is not the only one in which the shell plates cracked, and that several instances are known where this happened with plates that had been bent cold.

A very strong objection against bending shell plates while hot is the necessity of being possessed of a very long heating furnace, the extra time required for warming and then for cooling the plates, and the difficulty of obtaining a uniform temperature, which leads to irregular curvatures. These, and not the supposed injury done to the shell, are probably the most potent reasons which have induced manufacturers to adopt the plan of bending plates cold.

The Bending Rolls have necessarily to be proportionately stronger, and the following few notes will be a guide in the matter.

The resistance to bending beyond the limit of elasticity is independent of the curvature, and is approximately equal to $4 . t^2 . b$ for iron and $5 . t^2 . b$ for steel. Here t is the thickness and b the breadth of the plate, measured in inches. If the plate is heated to redness the coefficients 4 and 5 are reduced to about $\frac{1}{3}$ (see p. 145).

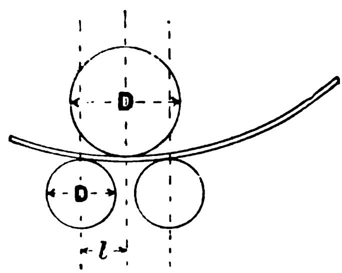


FIG. 215.

As the strength of the bending rolls is proportional to the cube of their diameters, and inversely proportional to the square of their length, and as the bending moment exerted by the rolls on the plate is, in the case of three rolls, of which the two smaller ones nearly touch (see fig. 215), proportional to the diameter, we find that $D^2 = c . L . t$ (see p. 133).

Here D is the diameter and L the length of the rolls, and c a constant which is equal to about $4\frac{1}{2}$ for the lower or outer ones of a system of three rolls, and the upper or inner one should be 25 % larger than these, as it supports double the load.

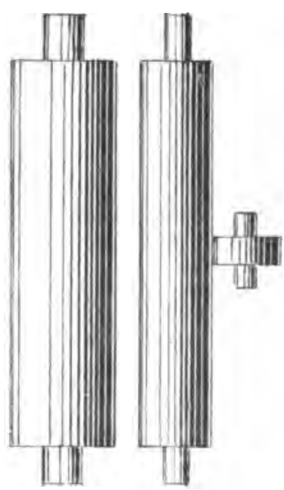


FIG. 216.

For red-hot plates the constants may be reduced to 1·5, which would allow of the rolls being reduced to about half the above diameters.

The smaller the diameters are made, the shorter will be the unbent end pieces of plate, and as this is a very desirable object, various devices are in use for attaining it.

The two outer (or lower) rolls are sometimes supported by anti-friction rollers, as shown in fig. 216. Or instead of three rolls four are used (fig. 217). The leverage x of the bending forces of the rolls can be very much reduced by this means, but on account of the greater pressure the diameters have to be proportionately increased.

Another plan is to have the four rolls arranged as shown in fig. 218, but the advantages are not apparent, and it would even seem that by removing the central roll C and bringing the two rolls B closer together the bending could be done better. In some machines the upper roll

can be moved horizontally, but this also demands that the diameters should be large.

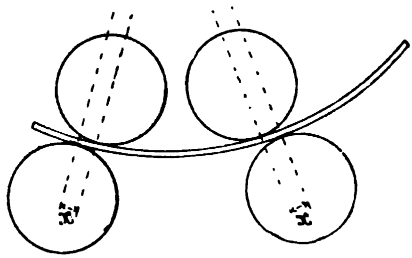


Fig. 217.

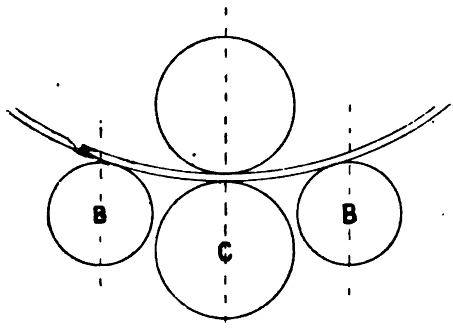


Fig. 218.

The bending rolls may be placed either horizontally (fig. 219) or vertically (fig. 220.) The latter plan is certainly the most convenient,

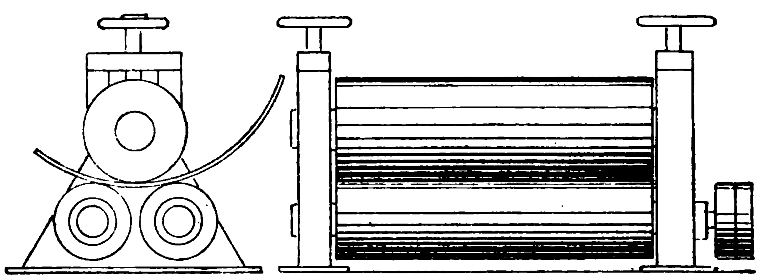


Fig. 219.

and is being generally adopted. The upper framework is shown in plan, and is so arranged that the inner roll can be lifted out, in order that shell or furnace plates may be rolled in one piece. Of the horizontal rolls it is usually only the two lower ones which are driven, while with the vertical rolls all three turn together. They thereby acquire a better grip of the plate, but even in that case it is advantageous to cut a few grooves into the driving rolls, as they materially assist in dragging in the plate.

Sketches of various types of bending rolls will be found in the following volumes of 'Engineering':—Horizontal plate bending rolls: vol. xxxiii. p. 134; vol. xlix. p. 529; vol. l. pp. 327, 480, 688. Vertical rolls: vol. xxxii. p. 135; vol. xliv. p. 258. Plate straightening machines with five rolls: vol. xl. pp. 9, 321, 619; vol. xlv. p. 135; vol. l. pp. 276, 606. Bending presses: vol. xliii. p. 491; vol. xlix. p. 245.

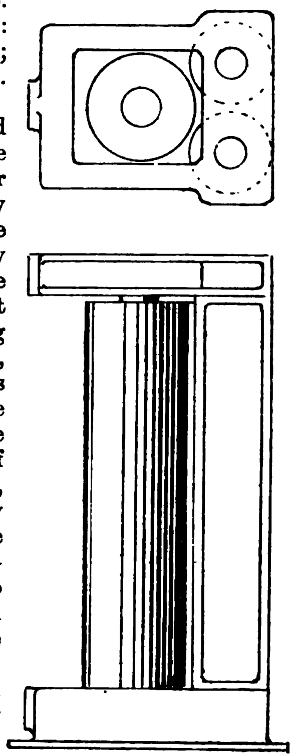


FIG. 220.

In some works the plates are passed several times through the bending rolls while these are being gradually screwed closer together. When possible, and particularly if the plates are hot, the curving should be carried out in one pass, for, independently of the disadvantages of punishing the material repeatedly, it will be found that less force is required for a single bending than for several, if that is necessary. This, however, is only possible if the machine is in good working order, and if full reliance may be placed on the marks to which the rolls are set. On account of the spring of the rolls some allowance has to be made, according as to whether very wide or very narrow plates are being bent, and for the same reason the influence of the extra stiffness of thick plates has also to be taken into account. After bending, the plates uncurl slightly, but absolute accuracy need not be aimed at.

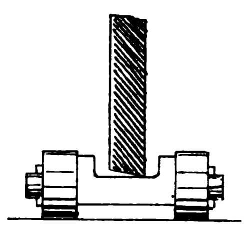


FIG. 221.

With vertical rolls the shop-floor should be square to their axis, and instead of using round iron rolling rods to support the plates, small carriages (fig. 221) will be found to follow the curvatures of the plate more smoothly, and not give rise to jerky motions. The speed at which the rolls are worked is about 18 ft. per minute, but it takes altogether about thirty minutes to bend one piece of shell plate. Before commencing the bending, a circular chalk-line, to judge of the curvature, is drawn on the floor, passing through the roll space; but it is also necessary to have curved templets, with which the upper edge is gauged, as the wear on the roller bearing is not an equal one.

Hydraulic Bending Presses are sometimes used instead of rolls. They seem to be most efficient for the bending of long narrow plates, particularly if the curvatures are all equal, as was the case with the tubes of the Forth Bridge. When the radii of the press moulds and the shell differ materially, liners have to be interposed, as in fig. 222, producing a rounded polygonal, instead of a perfectly circular shape.

This is the case even, though to a much less extent, when no liners are used, for the press is never powerful enough to force the plate into absolutely close contact with both moulds.

FIG. 222.

Templets have to be applied to the plates while being pressed step by step, otherwise irregularities are sure to occur. But no amount of care can do away with the irregular distribution of the stresses in the plates, and this plan cannot therefore be looked upon as a good one for furnaces, because with them the stresses are compressive.

A few sketches of some hydraulic bending presses are shown in figs. 223, 224. In the first of these the two rams act directly on the press frames, while in the other motion is imparted by means of wedges, W. In both cases it is necessary to let the two plungers move together; this is easily done by working the two force pumps from one shaft, and by having an accumulator with two rams instead of one.

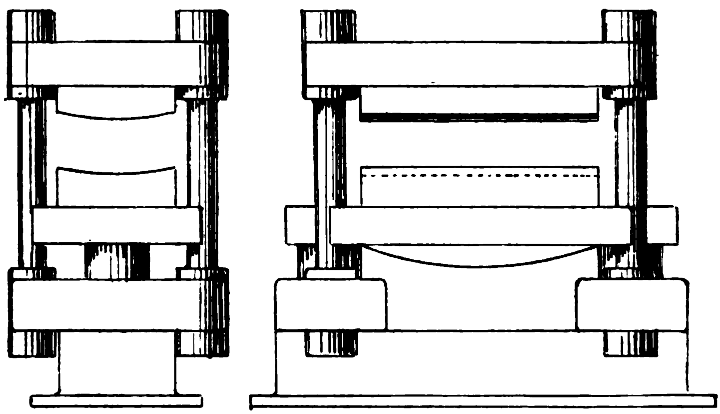


FIG. 223.

Shell Plate Ends.—From previous remarks it will have been gathered that one of the chief difficulties to contend with is the bending of the ends of the plates in such a manner that the general curvature is uniform, no matter whether the joint is to be butt-strapped, welded, or lap-jointed as in fig. 225.

There are various ways of producing these forms. In some works the end of the plate is heated before bending; it is then laid on a slanting anvil block (fig. 226), with or without curvature, and hammered with mallets. It is then heated over its entire length, and put through the rolls, or it is bent cold.

Another plan is to give the top or inner roll an extra screw-down when the end of the plate has been reached (fig. 227). In order to get the correct shape on the other end of a lap-jointed plate it has to be taken out of the rolls and reversed, as in fig. 228.

This plan is unsatisfactory unless carried out on hot plates, and

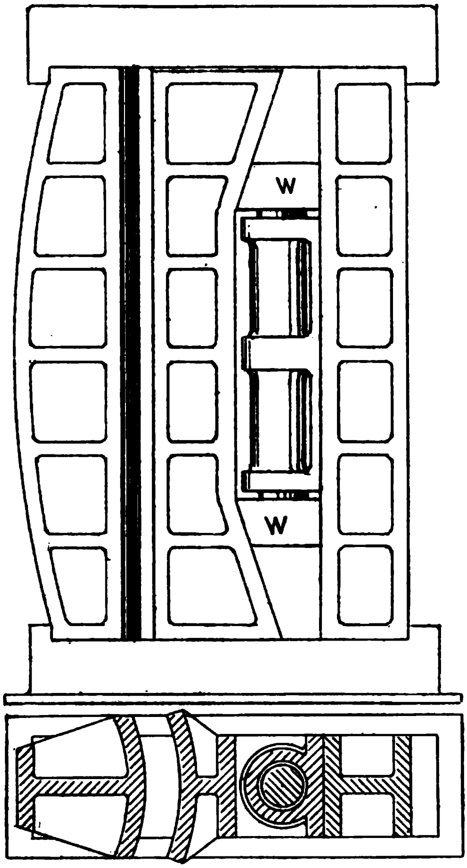


FIG. 224.

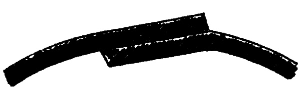


FIG. 225.

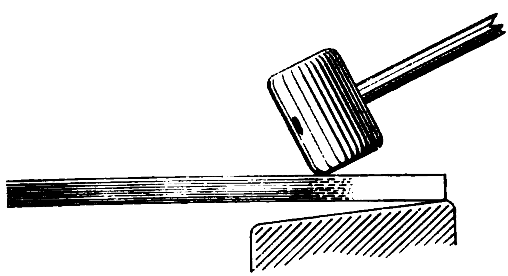


FIG. 226.

FIG. 227.

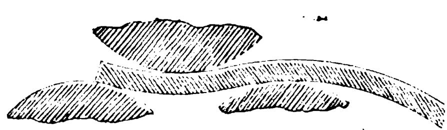


FIG. 228.

even then it is clear that the joining surfaces are not flat but irregularly curved.

A better shape is obtained by the following plan, but it is attended with great danger, because the bending may accidentally be carried out at a blue heat, producing either fracture or brittleness. The plate is fixed as shown (fig. 229), and a heater placed as at H, and when locally warmed the bending is done by hammering. The ends of plates to be butted are sometimes left flat, and are drawn together by the butt straps (fig. 230). Barbarous though this method is, the finished shape is a fairly true one, because the strength of the two butt straps is about equal to that of the shell plate, and the resultant curvature is the same as for the rest of the boiler. With thick plates this plan gives too much trouble while the riveting is going on, and under any circumstances it is not a satisfactory one. In some works the plates are ordered extra long and the ends cut off after bending and used as butt straps.

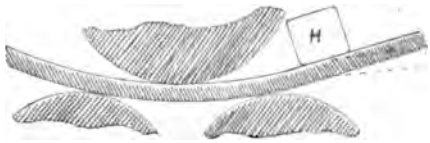


FIG. 229.

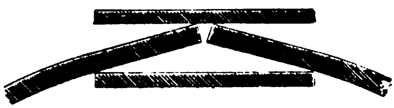


FIG. 230.

The most satisfactory results are undoubtedly obtained by bending the ends of the plates with suitable moulds in a strong riveting machine, either before or after bolting the plates together. The moulds (fig. 231) should be about 9 or 12 ins. long, and gently rounded at their ends. The bending or pressing is done cold. The same moulds can also be used for curving the butt straps.

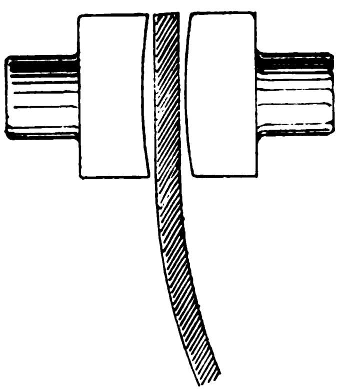


FIG. 231.

Ends of Riveted Seams.—In lap-jointed shell plates the corners have to be tapered off previous to bending (figs. 232, 233). This is usually done by heating them and drawing them out under a steam hammer or by hand. On account of the heaviness of the plates the latter plan is most convenient. Fin-shaped tools should be used, at least for the heavy work, and it is always well to re-heat the surroundings of such corners when finished to prevent cracking. Very satisfactory results are also obtained by chipping or planing the corners. (See figs. 234, 235, and p. 198.)

In the case of butt-strapped joints the arrangements are various. They are sometimes left square and butted at A A (figs. 236, 237) against the flanged end plates or the adjoining strakes, in which case it is very necessary to be careful that the lengths of the butt straps are correct, and also that they are correctly fitted. Another plan, but one in which the advantages claimed are doubtful, is to bevel the butt straps (fig. 238).

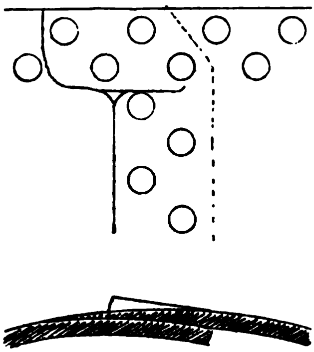


FIG. 232.

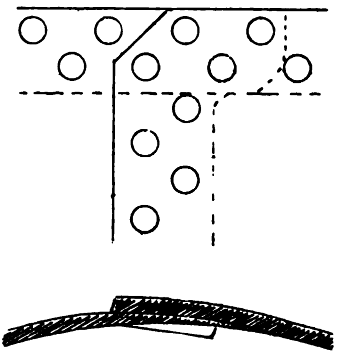


FIG. 233.

FIG. 234.

FIG. 235.

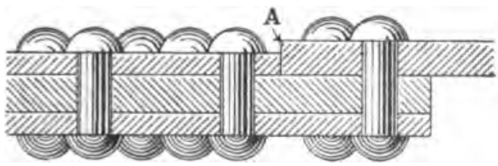


FIG. 236.

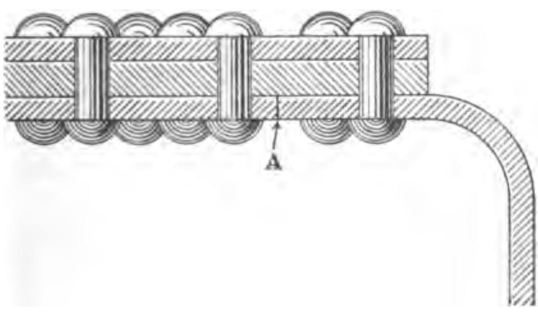


FIG. 237.

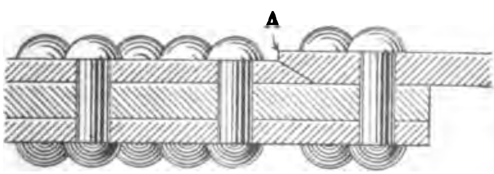


FIG. 238.

They are more difficult to make, to fit, and to caulk than the previous ones.

In both these cases, but especially in the first, it is customary to fit another cover plate, as shown in fig. 239.

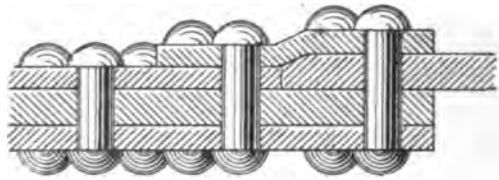


FIG. 239.

Another plan—probably the most efficient—is to draw out or plane the end of the outer butt strap and chip away part of the shell, as shown in fig. 240. Instead of chipping these parts they may with advantage

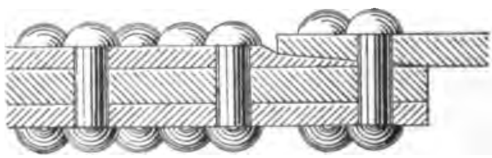


FIG. 240.

be planed before bending, and the butt straps might also be planed to shape. Fig. 241 shows another, but not a satisfactory, arrangement.

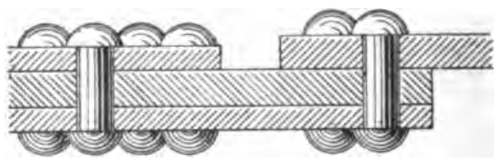


FIG. 241.

Sometimes the butt ends are welded, or the end rivet in a butt seam may with advantage be replaced by a screwed stud.

The Fitting together of the Shell Plates is comparatively simple, they being easily held together by temporary bolts and straps. With butt-strapped joints it is an advantage to be able to draw the butts tight

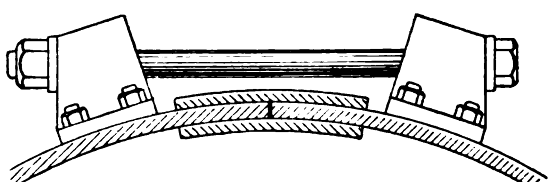


FIG. 242.

together, and this is best done as shown in fig. 242, by bolting brackets to the circumferential seams and drawing these together by means of strong

bolts. The end plates, either front or back, are made in two or three pieces, unless they are small enough to be made in one. The lower part of the end plate can now easily be secured in its correct position by bolts, starting at the bottom; this draws the shell up, and the upper part of the end plate can now be inserted and bolted to the top of the shell. Where the end plates are made of three pieces, the centre one is generally fitted last, but if it is desired to be prepared for having to renew the furnaces, it is the lower plate which should be fitted last.

Riveting.—The older types of power riveters were actuated by steam, while all newer ones are worked by hydraulic pressure. Formerly, too, it was thought necessary that the pressure should be applied suddenly, in imitation of the blows of a hammer; but it has been found that better results are obtained if the pressure is steady, provided, of course, that it is sufficiently intense. The smallness of the cylinders required for hydraulic presses has made it possible to introduce contrivances which will press the plates firmly together before and while the pressure is applied to the rivet. Machines have also been constructed which will form heads at both ends, so that, instead of rivets, pieces of round bars might be used. The results do not appear to have been very satisfactory, except that the heads thus formed required little or no caulking. Practically the same result can be obtained with an ordinary machine by using pan-headed rivets and a spherical die. The deformation which the head experiences assists in closing it up to the plate. Under any circumstances the rivets should be inserted from the inside of the boiler.

Riveting Machines.—Good illustrations of several types of riveters will be found in the following numbers of 'Engineering':—Vol. xxxiii. p. 199; vol. xxxv. p. 462; vol. xxxvi. p. 492; vol. xl. p. 317; vol. xlii. p. 80; vol. xlix. p. 259. Portable riveting machines: vol. xxxix. pp. 471, 474; vol. xliv. p. 289 (vol. xliv. p. 299, pneumatic); vol. xlix. p. 256. Hydraulic riveter with plate-closing arrangements, vol. xliii. p. 531. Hydraulic riveter for using rods instead of rivets, vol. xlix. p. 533. Portable riveter for the circumferential seams of boilers, vol. xliii. pp. 490, 491.

The riveting of the last circumferential seams can be performed by the last-mentioned tool when both end plates, having flanges turned inward, have been fitted in place. The front tube plate is made in three pieces, of which the centre one is not fitted till later; the boiler is laid on its back end, and the furnaces, which are not yet riveted to the fronts, are dropped to the bottom, and can, as occasion requires, be shifted to make room for the riveter. This consists of two long arms, of which one passes through the opening of the tube plate to the circumference; the other reaches the circumference from the outside, and contains the hydraulic cylinder, &c.

Another machine (fig. 243), designed for the same object, is worked by steam. To the centre (C) of the end plate is bolted a long arm with a powerful spring and a heavy holding-up weight (W), which presses against the rivet in the circumference of the end plate flange; a strong lever (not shown) is fitted, with which this weight can be pulled back and shifted to the next hole, where the rivet has previously been inserted from inside. The lever is now released, whereby

the rivet is pressed home ; a small steam hammer (H) is then made to strike the other end of the rivet, shaping it as required.

In works where either of these tools is used—where, therefore, both end plates can be riveted by machinery—the shell seams might

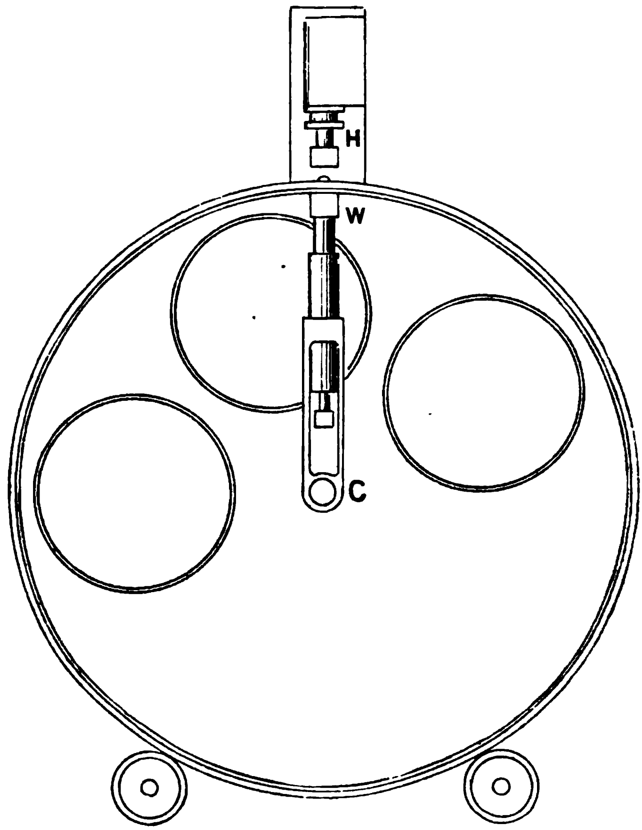


FIG. 243.

have been riveted with the help of comparatively light machines, similar to those used in the construction of the Forth Brigde.

Another plan, which enables all the circumferential seams to be riveted by machinery, is shown in fig. 244, which represents a section through the lower front end of a furnace and shell, the front plate having been flanged out.

Riveting Pressure.—Although machine riveting seems a simple matter, there are a few points which, if not attended to, will give trouble. The pressure used should not be too great, otherwise the plate may crack along the pitch line, either at once or when the boiler is in use. A pressure of 100 tons per square inch of section of rivet hole seems to be ample, while anything above 150 tons seems to

be dangerous. Allowing for the size of the rivet head, and assuming the red-hot rivet to be as fluid under pressure as water, this pressure would give rise to a tension of 50 tons per square inch in the metal surrounding the hole. Even allowing 50 % for friction and other causes, the stress is still an excessive one.

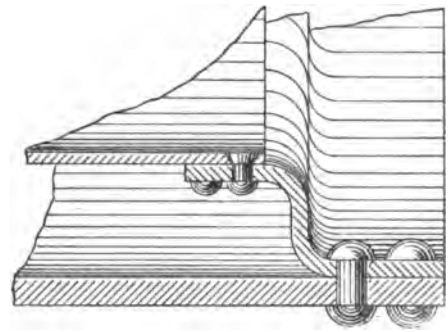


FIG. 244.

The distance of the metal from rivet hole to edge of plate must be sufficient to prevent bulging. On p. 165 it has been shown that, as regards strength of joint, it is necessary that the above dimension should be at least equal to $\frac{3}{4}\frac{p-n \cdot d}{N}$, where p is the pitch of rivets in the outer row, N the total number of rivets in one pitch, and n the number of rivets in the innermost row. This leads to the generally adopted practice of making this margin equal to the rivet diameter. The shearing stress which would be set up in this part of the seam, while riveting with a pressure of 150 tons per square inch of rivet hole, would be about 20 tons, which is also a dangerous amount, and the edges of nearly all machine-riveted butt straps will be found bulged.

It must not be forgotten that, as at present constructed, hydraulic riveters exert a greater pressure than their nominal one, for on opening the valve to the cylinder the accumulator weight falls, being arrested only when the rivet is struck. Of course the acquired energy of the drop makes itself suddenly felt as a very serious, but at present not measured, blow, exceeding by many tons the nominal pressure.

It is also important to see that the various plates are screwed close together ; otherwise, and particularly if the heads are small or countersunk and the rivets hot, plastic metal will force its way between the plates, as shown in fig. 245. Some riveting machines are so arranged that they can exert a pressure on the surrounding plates before and during the time that the rivet is being pressed. But whether this machine be used or not, it is always good to have the plates well bolted together : if possible, there should be one bolt in every alternate hole.

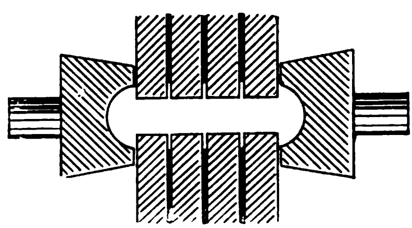


FIG. 245.

Bolting Rivet Seams.—The bolts may remain in place till all the alternate holes are riveted up, or they can be gradually removed while the holes are being successively filled. The latter plan seems to be both slower and more unsatisfactory than the first one, for it tends to stretch and shift the plates, and is adopted for this purpose in shipbuilding when the butts do not meet. It is claimed that this plan has

the advantage of warming the plate round the succeeding hole by the previous rivet, thereby preventing cracks; but such a danger ought not to exist with good material.

It will be found that the work can be done both better and quicker if the rivet holes are not filled up in succession, for, in order to do this, the bolts near the riveting machine have to be removed, and that, of course, is inconvenient. If the rivets are put singly into the holes, and at once riveted up, the boiler shell has to be shifted backwards and forwards over a distance of about 2 ft. before the next rivet can be inserted. If, on the other hand, three or four rivets are placed simultaneously into adjoining holes, the speed of riveting can certainly be increased, but that means that the second rivet is being pressed, while the first one is still hot and pliable, and it will almost certainly be stretched a little by the spring of the plate, so that some of the beneficial influences of the powerful hydraulic pressure are being wasted. On the other hand, if the pressure is kept up till the first rivet is cold there is no advantage in having three others waiting all this time in adjoining holes. This practice is illustrated in fig. 246. The rivet *d*

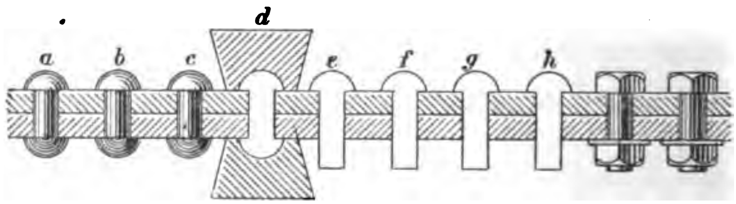


FIG. 246.

is being closed, and *e*, *f*, *g*, *h* are waiting while *a*, *b*, *c* are finished. In fig. 247 the rivet *b* is being subjected to the hydraulic pressure; *a* has just been pressed, but cannot well be affected by the new operation on account of a strong bolt filling the intermediate hole. The rivet *c* has just been placed in its hole, and can readily be acted upon when the pressure on *b* has lasted long enough. When every alternate hole has been filled, all the bolts are removed together, and their holes are

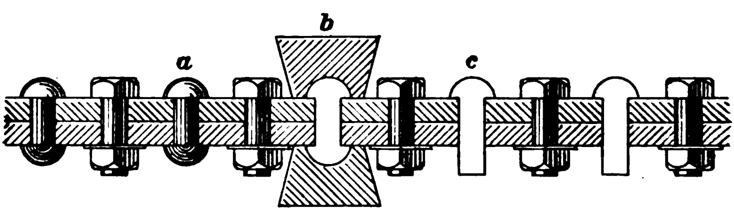


FIG. 247.

then riveted. After the bolts have been inserted, and before riveting is commenced, hydraulic pressure is applied all round the seam, and if this should slacken some of the bolts they are tightened up.

Irregularly-shaped Rivet Head.—Anybody watching the process of riveting must be struck with the primitive means used for guiding the rivet holes to the dies. An overhead crane, which might be more

usefully employed, is carrying the boiler, which is constantly swinging about. Crowbars are stuck into some of the rivet holes, and the men tug at these till the rivet, which has been inserted, is in the right position. As the weight of the boiler shell is often nearer 20 tons than 10 tons, it is natural that there must be much pulling and shoving till the right point is approximately reached. But even then the angular position of the shell is not correct, and the centre line of the riveter does not coincide with the axis of the rivet hole. Naturally, when the pressure is now applied, a few small but violent oscillations take place, and the chances are very much against the rivet head having been formed centrally round its shank. This is illustrated in fig. 248. Of course, externally there is no indication of this state of affairs, except, perhaps, that the dies leave semicircular marks round the heads, as shown in fig. 249. It ought not to be a difficult matter to devise appliances which would turn the boiler shell to its right position with more certainty than is at present possible.

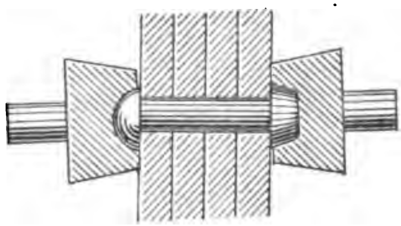


FIG. 248.

Time required for Riveting.—When the pressure has been applied it should not be taken off again in less than about one minute (and the longer it remains, the better); otherwise the rivet will not have cooled sufficiently, and the remaining spring in the plates will stretch it and reduce its diameter. The men do not generally care to keep the pressure on for so long, because the dies, &c., get hot and soft and soon wear out. It takes about 15 minutes to close up 10 rivets.

The Heating of the Rivets is done in a small reverberatory furnace. Sometimes gas or oil is used as fuel. They should not be raised to too high a temperature, or else they grow too plastic under hydraulic pressure and spread out between the plates (fig. 245, p. 195). Nor should they be heated too quickly, or else only their outsides are softened. On the other hand, it is not good to spend too much time over the heating, for a long exposure, particularly in an oxidising flame, reduces the strength of the material of the outer surface, and it is just this which is subjected to the severest stresses when under working condition. Fifteen to twenty minutes for heating is the general practice.

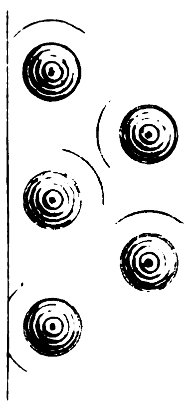


FIG. 249.

Unlike iron rivets, steel ones cannot be burnt without showing it when in place.

The influence of heating rivets to a red or to a white heat is discussed by M. Considère, 'An. Pont. Ch.,' 1885, 6th ser. vol. ix. p. 574, and 1886, 6th ser. vol. xi. p. 5.

Rivets are made about $\frac{1}{16}$ in. less in diameter than the holes they are intended to fill. The length, L, of their shanks has, therefore, to be

made greater than T, the combined thickness of the plates. $L = 1{\cdot}5\ D + T\ (1 + \frac{1}{8}\ .\ D)$, where D is the diameter of the rivet hole. The weights of rivets may be calculated by the formula

$$Q\ .\ \text{cwt.} = 2\ .\ n\ .\ \frac{D^2}{1000}\ .\ (L + 1{\cdot}6D),$$

where n is the number of rivets.

Internal Parts of Boilers.—Nearly all internal plates, as well as the end plates, have to be flanged. The only operations which precede this one are the bending and riveting or welding of the furnaces, the thinning or drawing out of some corners, and the drilling or punching of a few holes for temporarily securing the plates. Welding will be touched upon later, and the other preliminary operations require no remark, except, perhaps, that the drawing out of the corners may be done either by a steam hammer or by hand. The latter plan takes longer, but seems to produce a better job. (See p. 190.)

Before discussing the various flanging operations it is well to have a clear idea as to the risks attending them.

Dangers attending Flanging Operations.—Whenever a piece of iron or steel is being heated there is, of course, a danger of burning it. This should never happen while flanging, for during this operation such a heat ought not to be approached. A more real trouble is the wasting away which takes place, particularly with iron, which loses a considerable part of its thickness each time that it is re-heated. There is the further danger of reducing the thickness of the plate by drawing it out, and also the difficulty of producing the correct shape of flanges, and of preserving it, while adjoining parts are being heated.

More serious than any of these troubles is the risk of cracking the plates. There seem to be three ways of doing this:—1st, by using red-short iron or steel; 2nd, by not annealing the flanged plates, which then retain strains that may ultimately lead to ruptures; 3rd, the working of iron and steel at a blue heat. The plates which are thus injured will either break at once or, being now in a brittle condition, will crack later on.

On all these subjects interesting information will be found in the chapter on 'Strength of Materials.'

The Danger of not Annealing flanged plates has been demonstrated over and over again in boiler yards by plates cracking. Few of the makers care to give details of their experience, especially as the steel manufacturers readily substitute new plates for the spoilt ones, and do not care to have such cases noised abroad. Those which have been published will be found in the chapter on 'Strength of Materials,' but numerous others are continually occurring. Flanged tube plates (figs. 250, 253, 256), flanged end plates (fig. 255), furnace fronts (fig. 251), and with ships the huckster plates, boss plates (fig. 254), and garboard strakes, sometimes give trouble by cracking.

Stresses Due to Flanging.—Here it will not be out of place to draw attention to the fact, which is thoroughly supported by experiments, that iron and steel are affected in their quality by severe stresses, and particularly that, by subjecting test pieces to a slowly increasing but severe pull, their ultimate strength can be raised, while their ductility is very much reduced. The most important point is, that the limit of

elasticity rises under this stress, until it actually exceeds the original strength of the material. In other words, taking a test bar whose limit of elasticity is 10 tons, and whose ultimate strength is 30 tons, with 20 % elongation, it is possible, by slowly increasing the load, to raise both the limit of elasticity and the ultimate strength to about 35 tons, while the elongation which takes place during the final loading is reduced to a very few per cent., possibly to nothing. When this point is reached, the test piece gives way completely, without any additional weights being added. Now this gradual increasing of stresses is reproduced while flanging a plate. The shape does not matter much, except, perhaps, as regards the intensity of the stresses which are set up. In forming a tube plate, first one side is flanged, except at the corners, then the next one in the same way, and then the third. When this is finished the corners are once more heated and flanged. Now it is clear that, if the operation commences at A (fig. 252), the heat will have little effect at first ; but even while B is being flanged and A is cooling, strong tension stresses would be produced ; these are very much increased by the time that C is finished. Each one of these parts has probably had a chance of cooling slowly from about 1,000° F., and if the plate were now put aside, and measurements could be taken, it would be found that these lengths of the plate had not shortened as much as would be due to their change of temperature. If a steel bar, 1 in. in area, is rigidly secured at its ends, and cooled from 212° F. to 32° F., it would then exert a pull of about 13 tons, so that it is not unreasonable to imagine that the cooling of the flange from a red heat would produce a stress of at least 20 tons. But a tension stress at one edge of a plate must produce a compression stress at the centre and a tension at the circumference. In other words, the plate yields a little,

FIG. 250.

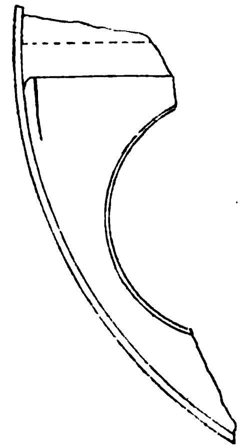


FIG. 251.

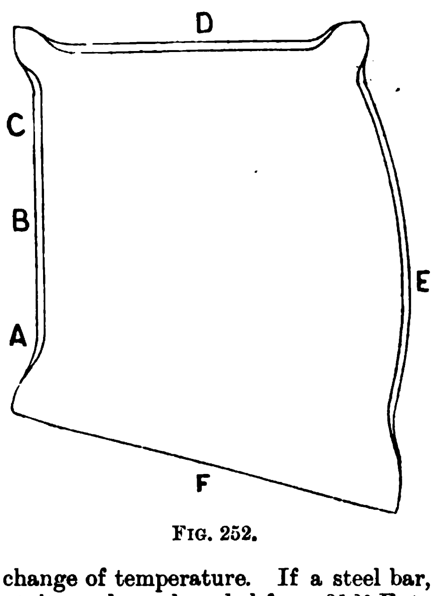


FIG. 252.

and possibly, instead of finding a stress of 20 tons at only one edge, we find a fairly uniform stress of 10 tons all round the plate, and a compression stress in the centre. While the second side is being flanged new stresses are set up all round; then, when the third side goes through this manipulation, the stresses will be once more increased. To give a correct idea of the distribution of the stresses would be impossible, but for the purpose of illustration it may be assumed that the flange ABC is now subjected to 30 tons tension, the flange D to 20 tons, and E to 10 tons. The unflanged parts, F, would be subjected to 30 tons, and the centre of the plate is in compression. The next operation consists in flanging the corners, and there is no doubt that heating them relieves the other stresses at the circumference a little, but on cooling they will be more intense than before. Even now, if the cooling were carried out quickly, there might be no danger, because one or the other part would gently elongate; but, as is usually the case where a failure has subsequently occurred, the plate has been put aside for the night, and next morning the flat part had cracked (fig. 253). Slowly the breaking stress was reached, and then, as in the case of test pieces, the material gave way completely. In a case mentioned by the late Dr. Kirk (fig. 253), 'N. A.,' 1882, vol. xxiii. p. 131, the crack extended about 18 ins. into the plate, and measured $\frac{1}{2}$ in. open at the edge. It is not at all certain that, by cracking, the plate was relieved of all its strains; but even assuming this, and assuming also that the stresses were uniformly distributed over the total circumference of about 14 ft., the opening of the crack to $\frac{1}{2}$ in. would show that in this particular case the circumferential stress amounted to $\frac{1}{2} \cdot \frac{13,000}{168}$, or nearly 40 tons per square in., showing that the above estimate of 30 tons was not too high. Other cases are illustrated in figs. 250, 251, 253, 254, 255, 256. It has not yet been possible to obtain tensile tests of plates which have not failed, but which might be expected to do so. They would certainly throw a strong light on the subject. Test pieces should be sawn (not sheared) out of the plate all along its circumference. An accurate measurement of the limit of elasticity would give conclusive information as to the intensity of the stress which that particular part had been resisting; but the greatest care would have to be taken not to bend the sample (p. 121).

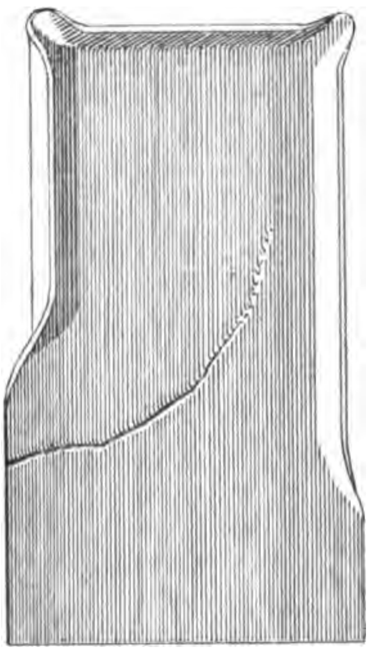


FIG. 253.

Before leaving this subject it may be as well to explain why cracks

of this sort extend so far beyond the overstrained part. The contraction of area at the edges of the crack at its inner end, and even beyond this point, is a sure sign of the ductility of the metal, at least in the centre of the plate. A careful study of the subject will show that at the

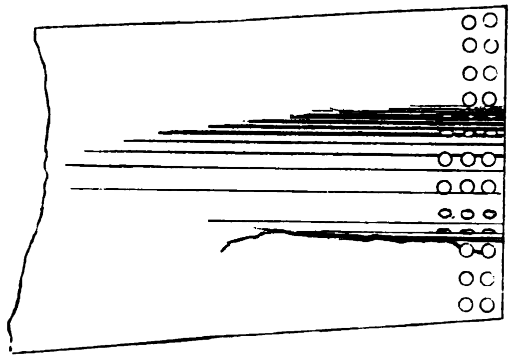


FIG. 254.

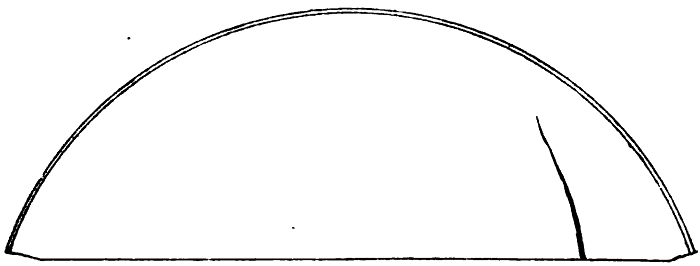


FIG. 255.

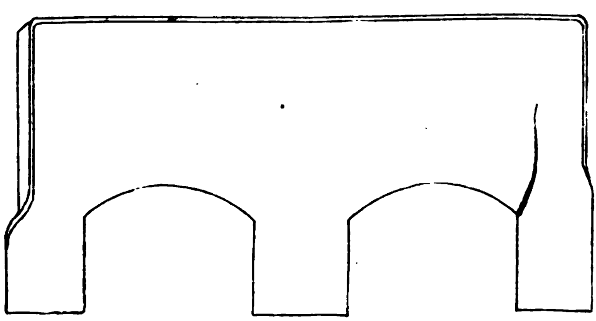


FIG. 256.

instant when a fracture takes place the two separating surfaces are travelling away from each other at a velocity equal to that of sound. For iron and steel it is 17,500 feet per second. The rate at which the crack mentioned above was extending towards the centre of the plate must have been = 1,260,000 feet, or 550 miles, per second, and it is

therefore not surprising that it had overshot its mark. Instances could be mentioned where the crack had extended right across the plate, and in one case a plate actually broke in two, one piece knocking down a man. If the above estimate of 40 tons stress per square inch is a correct one, then the amount of energy which was relieved by the plate cracking was equal to lifting it bodily 16 feet into the air. The loudness of the report when these cracks occur, and the destructive energy which some steel armour plates have displayed when cracking spontaneously, are proof that these estimates are not excessive. The existence of such mischievous powers in the interior of boiler plates should not be tolerated, and there ought really to be no objection to the annealing of plates which have been flanged. When this cannot be done at once, the centre of the plate should be heated to redness immediately after flanging, and before the edges have lost their heat. The compression stresses in the centre of the plate are thereby partly removed. If the plates have been very much buckled during flanging, they should not be flattened except during the annealing process, for it is chiefly the very flat plates which crack.

Working Plates when Partly Cooled.—The other danger to which allusion has been made is the working of steel and iron at a blue heat. Experiments on this subject will be found in the chapter on 'Strength of Materials.' Here it is only necessary to draw attention to workshop practices. The ease with which such failures can be attributed either to redshortness, coldshortness, or generally unsuitable material, and the absence of any chemical or mechanical test, make it difficult to be sure, in any particular case, what has been the cause of the breakage; but about the following one there can be little doubt.

The flange E (fig. 257) of part of a furnace front plate was so much out of shape that the furnace, *f*, could not be drawn up sufficiently to make a good job, and it had to be knocked in a little; but being rather thick, a substantial heater, H, was first applied, as shown. A few weeks later, while the boiler was being tested by hydraulic pressure, a loud report was heard, and though no leakage took place an internal crack, *c*, was ultimately discovered, extending round one-quarter of the flange, yet penetrating only to within one-sixteenth of the outside surface. Too little is as yet known about this subject, though every boiler-maker should be made aware of it, which can easily be done by letting him make the following bends:—

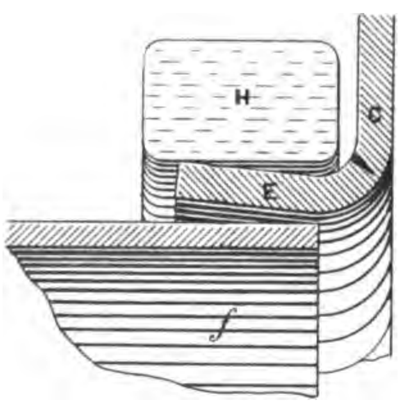


FIG. 257.

Strips of mild steel 6 ins. long, 1½ in. wide, and about ⅜ in. thick, should be treated as follows:—

No. 1 should be placed under a steam hammer (fig. 258), allowing about 3 ins. to project as far as *a*. This end should then be bent down to the angle *b* by striking it with a sledge hammer. It should then be

reversed to the position *c*, and again bent down, and the operation continued till breakage takes place. If good, the material will stand 20 half-bends, either with sheared edges, annealed, or hardened.

No. 2 should be placed between two heaters, H, H (fig. 259), and kept there till its edge turns straw colour to violet. It should then be taken to the steam hammer and bent as before. Two instead of 20 bends will now suffice to break it.

No. 3 should be treated like No. 2, but the bending should only be carried on till there is the first indication of a crack. The sample should then be put aside for a day to cool slowly, when it can readily be broken with a hand hammer or by throwing it on an anvil.

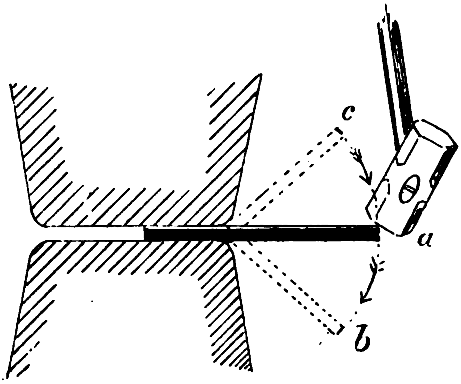


FIG. 258.

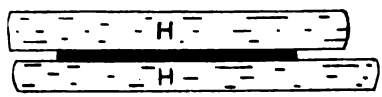


FIG. 259.

No. 4 should be heated like No. 2, and then drawn out under the steam hammer till its thickness is reduced by about $\frac{1}{16}$ in. After waiting a day it will have become as brittle as glass.

Cases in which the influence of working flanges at a blue heat may at least be suspected are shown in figs. 250, 251.

Local Heating.—It may be possible that plates can be injured by the influence of a blue heat even while they are partly red-hot, and that is another reason why they should always be annealed after working them.

Fig. 260 shows a plate partly flanged, and also locally heated near the centre of one edge. Evidently this red-hot part is surrounded by a

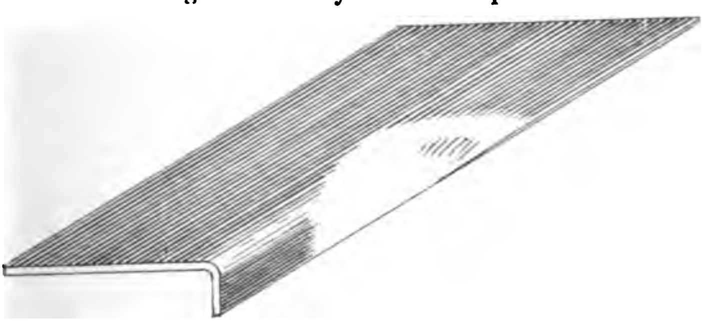


FIG. 260.

zone of which one part is blue hot. There is also a strong probability that one part of this zone will retain its temperature for a long while,

for although the whole plate is radiating heat, the colder parts do not lose it as quickly as the hot ones, and are also being warmed by them, so that at some point the gain and the loss must be equal during a considerable period. If the plate is being hammered while this point of permanent temperature is just blue hot, very serious injury may be done to the material. It is therefore not as improbable as it might otherwise seem to find that such a plate contains as many brittle zones as there have been heats applied to it. The danger of these local defects is accentuated by the adjoining parts being tough and ductile, and such a plate might be compared to one made up of glass ribs and sheets of lead. There is naturally much difficulty in reproducing these conditions experimentally, but the danger exists all the same, and should be guarded against by careful annealing.

Cases have occurred where unflanged plates whose corners had been drawn out and then laid aside cracked overnight. It is difficult to imagine that the quality had nothing to do with this, though the tests were good, but stresses had evidently been set up in a similar manner to those explained on p. 199.

Hydraulic Flanging. Formerly presses similar to that shown at fig. 223, p. 188, were used for flanging, but the necessity of requiring numerous moulds and various other inconveniences has led to the very extensive adoption of Tweddell's flanging press (fig. 261). It consists of

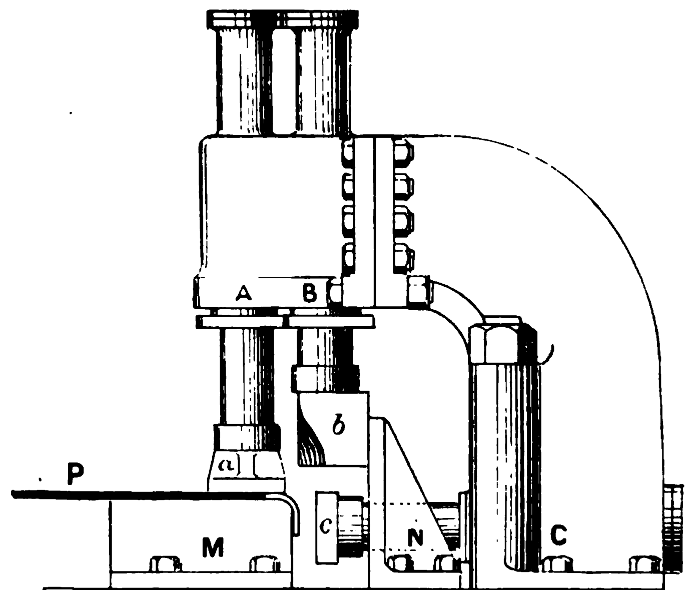


FIG. 261.

three hydraulic cylinders, A, B, C, to whose rams various moulds or head-pieces can be keyed, *a*, *b*, *c*. The whole is supported on a strong bed-plate, B, to which the mould block M and the angle frame N (for

guiding b) can be bolted. The press is shown in the act of flanging a boiler end plate, P, which is firmly held down by a, while the moulding iron b is forced down the side and bends the plate. After each stroke a is lifted so as to allow the plate P to be moved a little, and then b descends again. When the whole length of one heat has been dealt with in this way, the flange will be very irregular and frilled, as shown in fig. 262. These irregularities are removed by forcing the ram c against the circumference of the flange. The head-piece a should be of ample size, so as not to injure the plate, but not too large, otherwise the plate remains perfectly flat after flanging, in which condition it is more liable to crack than if somewhat warped or buckled. The plate should be hottest at the edge, for if that part is left dark red, the thickness of the plate will be reduced at the bend, and the puckers cannot be easily removed. Fig. 263 shows an end plate partly flanged. One part, a, is bent down, the other part, b, is still straight, and the metal between a and b must have stretched considerably. Besides this the circumference of the plate at b is greater than that of the flange a, so that when finished there must be a considerable compression stress in the flange. It has the effect of slightly bending the plate edgeways, as can be noticed by the curving of any straight line which has been scribed on the plates before flanging.

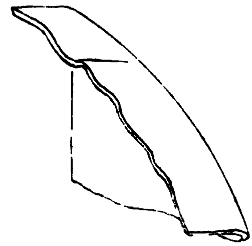


FIG. 262.

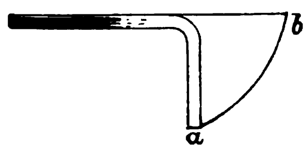


FIG. 263.

Short lengths, varying from 4 to 6 feet of the circumference (and even 8 ft. of thick plates), are heated at one time and the flanging completed before the next length is taken. The heating takes about twenty minutes, and the flanging is at the rate of about 1 foot in 2 minutes. The extreme ends of the flanges are never of the shape finally required, and have to be dealt with subsequently by hand.

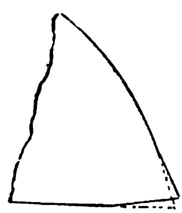


FIG. 264.

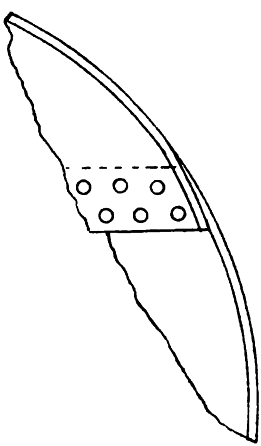


FIG. 265.

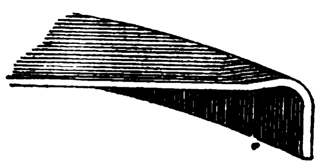


FIG. 266.

Fig. 264 shows how the corners of the plates are cut previous to flanging. The outside plate (see fig. 265) is drawn out at the corner,

and when subsequently flanged has a very irregular appearance, as shown in fig. 280, p. 212. Fig. 266 shows how the corners rise up. This is due to the above-mentioned compression of the material. Before allowing the flange to cool, the corners of the inside flange have to be knocked in with a few blows of a hammer, and the corners of the outside flange knocked out ; otherwise these parts will have to be re-heated before fitting together, which would be a great waste of time.

Flanging with a Steam Hammer is done when no other means are available. The plate is placed at an angle on rollers (fig. 267). H is the moulded hammer-head ; M is the lower mould, keyed to the anvil block. A tie rod or plate is also bolted to it for holding the pivot of the plate P, which has been firmly bolted to the centre. The flanges occasionally get torn by this rough treatment, and must then be welded.

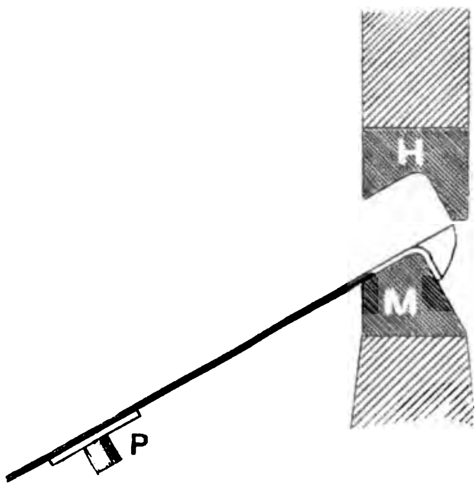


FIG. 267.

Hand Flanging.—It goes without saying that up to certain thicknesses flanging can be done by hand. It is usual to dispense with pivots and stops, which are necessary with machine flanging, and to be guided as regards shape by deep centre punch marks on the plate. For end plates it is always better to use a pivot ; the circumference can then be made more truly circular.

The size to which plates have to be sheared for flanging depends, of course, on the depth of the flange, but also on the skill of the operator. With some a very much greater margin must be left for irregular work than with others. The following is a customary rule, and produces flanges with an average of 1 inch for waste :—

To the external size of the flanged plate add the depth of the flange, measured inside.

For end plates of boilers this is the usual custom. The furnace holes are, however, cut 2 inches smaller in diameter than would be required by this rule, while for machine-pressed dome ends the extra margin for the flanges is reduced to about 75 % of the above, because they draw out considerably.

Tube plates and combustion chamber back plates have to be ordered with a margin which is 1 inch in excess of the depth of the flange.

An extra allowance of 1 inch has to be added at the corners of plates to be flanged by a steam hammer. (See fig. 264.)

Flanging Furnace Holes.—Another operation which is carried out under the press is the flanging of the holes in the furnace front plates. Having been bored out to the correct diameter, as indicated above, the

circumference of each hole is heated and flanged separately. This is also done under a Tweddell's flanging press. A strong cast-iron ring mould is bolted to the bed plate, and the two plungers *a* and *b* (fig. 261) are secured to a strong cast-iron die. The cocks and valves of the press are then altered, so that the plungers work in unison, and when the furnace front plate has been placed in its proper position on the ring mould, the die is forced through it, producing the desired flange. The power required seems to be at the rate of about 5 tons per foot of circumference with a $\frac{3}{4}$-in. plate and with a sufficient depth for one row of rivets, and double this power for inch plates or for treble-riveted flanges. After being heated it takes about 15 minutes to carry out the flanging.

The furnace front plate should be firmly held in its proper position, because a tendency exists for the die to drag down only one side of the flange, and in doing this the plate gets moved. If the ring mould is strong enough, the plunger *c* might secure the plate. The die should also be as taper as the length of the stroke will allow ; less force is then required and the tendency to shift the plate is reduced to a minimum.

Irregular Shapes of Furnace Holes.—Simple as the operation appears, it will be found that the holes produced in this way do not always turn out perfectly circular, unless certain precautions have been taken. One of these is not to heat the plate further than is absolutely necessary, so as not to warm, and thereby weaken, those flanges which have already been finished. Thus, if the finished flange at *a* (fig. 268)

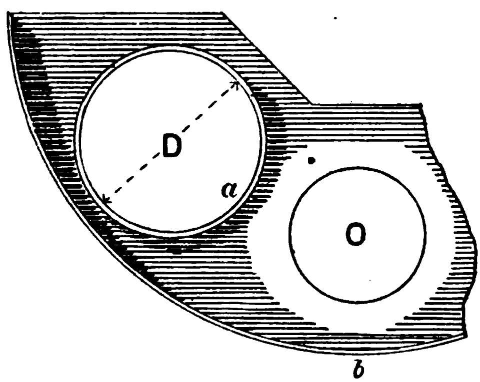


FIG. 268

were to be heated while the circumference of the hole O is over the furnace, a slight contraction would take place at this point during flanging, and the diameter D would be reduced. Similarly, a contraction would take place at *b* if the outside flange were heated at this point. This contraction is the necessary consequence of the stretching of every part of the circumference of O, just as the closing-up action, while flanging the outer circumference of end plates, tends to elongate the adjoining parts.

These deformations can be partly prevented by placing a stout iron

ring (fig. 269), cut as shown, inside the finished flange and tightly wedging it into position, but only after the adjoining hole has been heated,

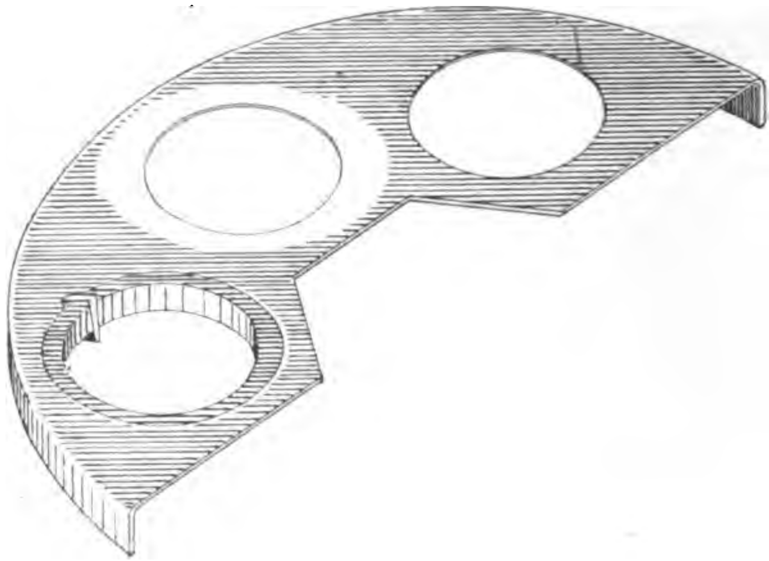


FIG. 269.

and just before it is placed under the press. In some works all three or four furnace holes are flanged in one operation, and sometimes also the outside flange is done at the same time. This of course requires a very powerful press, similar to the one shown at fig. 223, p. 188. The great inconvenience of this method is the large number of moulds and dies required; but the results are highly satisfactory, the only serious trouble being the drawing away of the metal from the weaker or lower flange towards the stronger one (see fig. 270), which very often leads to the outer flange being higher and the inner one being lower than either should be. Fig. 271 is a section through a furnace-hole flange, and shows the deformation to be expected.

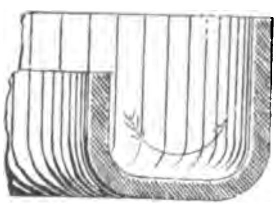


FIG. 270.

The actual flanging operation is easier with iron than with steel, because this metal is softer, and because it can be worked at a higher

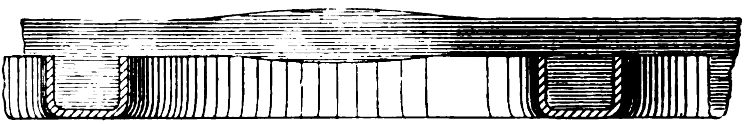


FIG. 271.

temperature, and fewer heats are necessary. On the other hand, only the very best qualities of iron can pass through this operation without

showing cracks or other defects, and there is also much waste of thickness, due to the burning of the surface. That steel possesses great advantages is proved by the fact that hardly any other material is now used for this purpose.

Annealing.—It might have been better to postpone the necessary remarks about annealing until all flanging operations have been discussed, but, as there is no more difficult piece to deal with than a large furnace front, the subject has been introduced here.

The heating furnace is of the ordinary reverberatory type (fig. 272). It is usual to employ coal for fuel, but gas is also used, and it has been

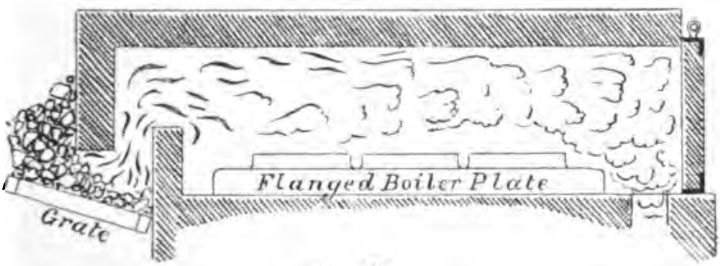


FIG. 272.

affirmed that it causes plates to grow brittle. The flame travels right across the furnace, and gets drawn down at the front. If the furnace is very large, the firing is done at one side, and the downcast is placed at the other. Doors should be fitted so that the plate can be watched, and care should be taken to keep the temperature comparatively low, partly in order not to burn the iron or steel, but particularly not to heat one part of the plate before another ; otherwise serious distortions will occur. Large plates are taken out, turned end for end, and replaced before finally allowing them to cool.

Deformations Produced by Annealing.—Even with the most uniform heating it will be found that the strains which have been produced while flanging will make themselves felt in the most annoying manner. As already mentioned, the tilting up of the outside flange shortens it, so that it is in compression, and the adjoining part of the plate will be in tension. In a furnace front plate this is counteracted by the stretching of the furnace-hole flanges, but only partially ; for, on heating such a plate, it draws as indicated in fig. 273. The metal at *a*, *b*, and *c* elongates, and the outer flange comes in, sometimes as much as ½ in.

The metal shortens at *d* and *e*, so that in one direction the diameters increase and in the other they decrease, their difference occasionally amounting to as much as 1 in. Partial relief is given by annealing the furnace front plate after the circumference has been flanged, and once more when the holes are finished.

To flange the furnace holes oval, which is sometimes done, does not give good results, partly because the changes of form cannot be previously estimated, but more particularly because the outside flange gets drawn in during annealing, and then, although the furnace holes may be circular, the outside flange is locally flattened.

All these distortions depend not only on the shapes and sizes of the flanges, but also on the nature and number of the heat at which the various parts were bent. By making the outer flange and that of

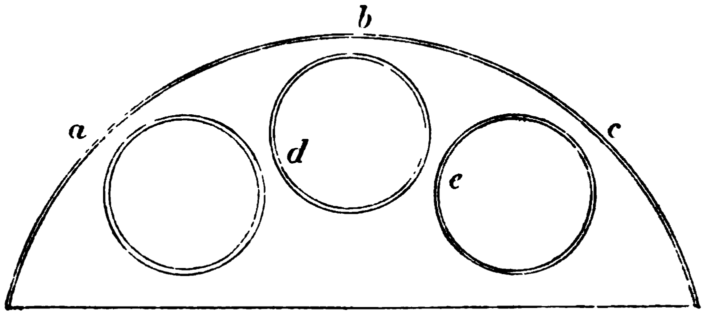


FIG. 273.

the furnace hole perfectly circular, then reheating both locally, and forcing them out at this point by means of a liner shown in black in fig. 274, and then annealing the plate, the flanges return from the

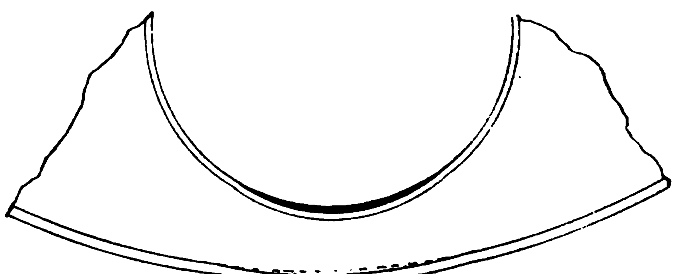


FIG. 274.

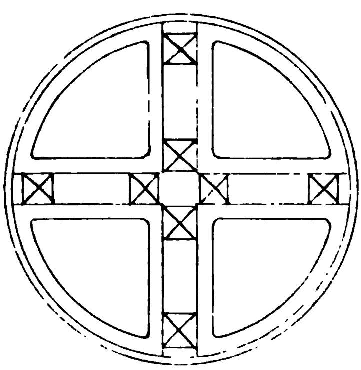


FIG. 275.

position shown by the black line to their original shape, as shown by a dotted line. If they are still oval these parts are set by hand during the process of annealing, the plate being drawn out of the furnace for this purpose and partly flattened, and then replaced and reheated, and if necessary reset. Where great accuracy is aimed at, solid segments of circles (fig. 275) are wedged into the furnace holes before annealing; they have to be made rather strong, but for all that waste away if used too often.

A careful examination of machine-flanged holes, after they have been annealed, will show that they

have closed in a little at their edge (fig. 276). This, like the other deformations, is due to the stresses set up by the flanging operations. It is needless to say that all other flanged plates are distorted more or less by annealing. End plates cannot, therefore, be fitted together before this is done.

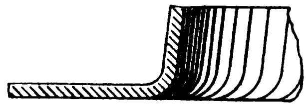


FIG. 276.

The plates are flattened when they leave the annealing furnace; this is very necessary, especially along the edges, where the riveted seams come. The operation is not difficult, but care must be taken that the necessary hammering of the flanges does not spoil them. It is necessary to test the circumference by templet while the plates are still hot, and to gauge the furnace holes. Occasionally the circumferential flange will be found set up as in fig. 277. This

FIG. 277.

happens particularly near the furnace holes. In such cases heavy double-handed hammers have to be used to knock it down again. Portable steam hammers are also used for this and other purposes.

Plates in which the flanges of the edges and of furnace holes are turned towards opposite sides are troublesome objects. Not only is it difficult to get them level, but at the points where the two flanges are nearest each other a disagreeable tendency exists for them to tilt over (fig. 278), and also to bend down bodily, as shown in fig. 279. Considerable experience is necessary to get these parts into shape.

Plates should remain in an annealing furnace for at least two hours, but it takes a few more hours before they have cooled after being laid

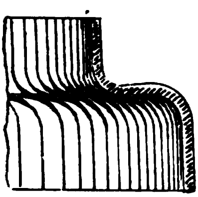


FIG. 278.

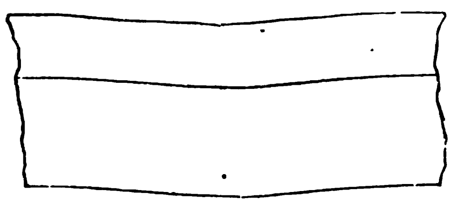


FIG. 279.

on the shop floor. A lesson in annealing may be learnt from the plan adopted for optical glass, which leads to perfect results. It is heated to the required temperature, and the furnace is then cooled quickly, but only through a small range of temperature, and it is only when the whole of the glass has adapted itself to the reduced heat that it is lowered once more. This is repeated till ordinary temperatures are reached. If the temperature is reduced steadily the surfaces would necessarily be slightly colder than the inside, and straining could not be prevented.

Final Flangings.—Having been thoroughly annealed, the various parts of the end plates are now clamped together, care being taken

that the circumference is a true circle, and of the right diameter, so as to fit the shell. A few holes are drilled through the various cross-seams, and these are then screwed together by well-fitting bolts. Girders, or, for small-sized plates, angle irons, are also bolted to the plates, to keep them flat while suspended, and then one corner joint after another is heated and brought to its correct shape by hand hammering. This is rather a troublesome operation, and generally requires two heats, but if the corners are carefully prepared much time is saved. When, finally, the two plates are in close contact at the corners templets are applied, to test whether the curvature is correct. These parts should project slightly, as they get drawn in on cooling.

The shaping of these corners will be best understood by referring to fig. 280, which shows the irregular shape of the corner plates as they have left the press. The corner is now heated, and the outer plate struck, as shown in fig. 281, starting at the bottom, so that the humps

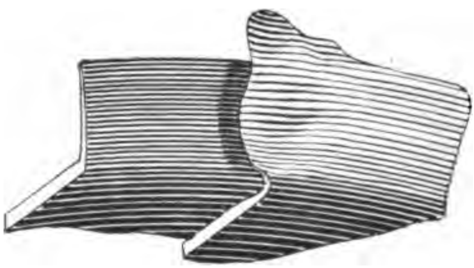


FIG. 280.

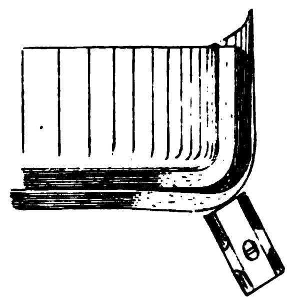


FIG. 281.

(which are shown in fig. 280) of both the outside and inside plate are driven in. Gradually the upper parts of the flanges are reached by the hammer. In the meantime, especially for light plates, heavy hammers or weights have been held inside the flange, in order to make the blows more effective and to bring the plates well together. At a later stage the inside is struck to drive that flange out and to stretch the outer one a little. All this hammering is performed with the end plates, suspended by a chain, so that the heated corner is quite accessible. The final operation is performed on a slab of iron (fig. 282). If there are two horizontal seams in the back end plates, it is usual to rivet up one of them, either before or immediately after the above operation. In shops where these corners are welded (fig. 283) none of the seams may be riveted up first; the shaping of the corners is done during and immediately after welding. After this operation it is well, and even necessary, to reheat the neighbourhood of these parts. This has to do service instead

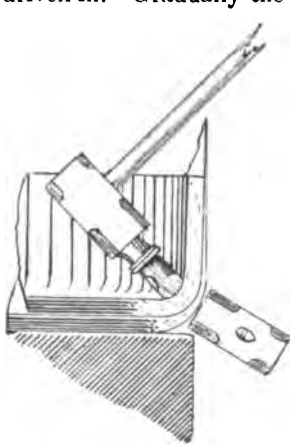


FIG. 282.

of annealing. These welded corners contract about $\frac{1}{8}$ in., and it is necessary to make the proper allowances. It would be well to leave these corners full and trim them cold. The stay holes, unless they have been punched before annealing, or if it is intended to bore them in place, are now drilled, and the plates can now be fitted into the shells, drilled, and riveted up, as already explained. In some works the straight edges of all flat plates are bent so as to make bevelled joints; but this is unnecessary, costly, and dangerous, for heaters have often to be used, and the plates may thus be made brittle.

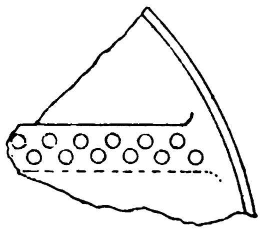


FIG. 283.

Flanging Furnace Saddles.—Undoubtedly, one of the most difficult flanging operations is the shaping of furnace saddles. Recent attempts to do the work by Tweddell's flanging press are said to be highly satisfactory.

Two objects have to be kept in view, viz. not to burn or otherwise injure the material, and not to reduce its thickness. Suppose that a cylindrical shell (fig. 284) is marked with a number of equidistant longitudinal lines, and that the end of this cylinder is flanged like a furnace saddle, as shown in dotted lines, the straight ones can be made to take up any of the positions shown in the end views (figs. 285, 286, 287). In the first case the lines radiate from the furnace centre, and it is clear that the metal will have been much stretched, and con-

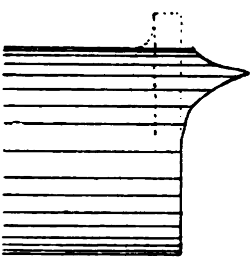


FIG. 284.

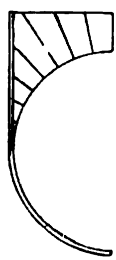


FIG. 285.

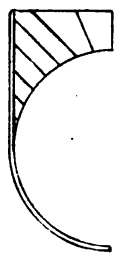


FIG. 286.

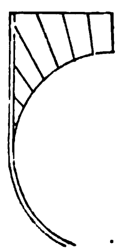


FIG. 287.

sequently much thinned, at the corner. This happens if the corner is made very hot during flanging, and is also flanged first. In the second case the lines which extend up to the corners are parallel, showing that thus far the metal has not been stretched or thinned. But that is not the case from the corner to the crown, where the lines spread out very much. This can only happen if the centre has been heated and flanged first, and kept hot while flanging the corners. Fig. 287 shows how the lines should have distributed themselves. The available means for attaining this end are: firstly, not to heat the whole width of the flange, but only the curved part, leaving the edges of a dull red heat; secondly, to do the flanging gradually, and not to finish one corner outright, but to partly flange first one corner, then the other, and so on; thirdly, by leaving an extra amount of metal on those parts of the flanges which are most easily drawn thin, and then if their edges are not heated too much they might even thicken some parts near the curve by stumping them up.

The most difficult furnaces to flange are those with high saddles, as shown in figs. 288 and 289. As nothing is gained by making them

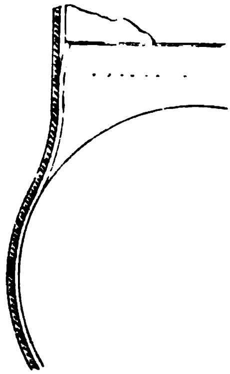


Fig. 288.

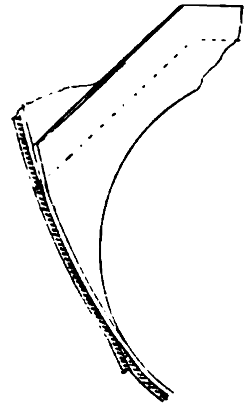


Fig. 289.

of this shape, it is better to keep the flanges as low as possible, as shown in figs. 290 and 291. The deep flanges usually require from

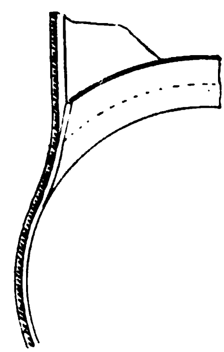


Fig. 290.

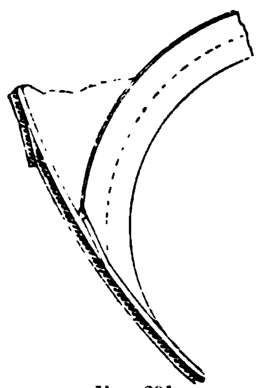


Fig. 291.

fifteen to twenty heats, while it is asserted that some smiths can produce the low flanged saddles in five or seven heats, but in that case the curvatures must be kept large. Each heat requires about one hour.

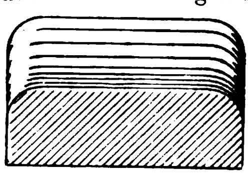


Fig. 292.

The Flanging Operation is carried out as follows :—The anvil mould M is shaped as shown in section (fig. 292). It is part of a segment of a circle, and is bolted to a strong bed plate, B (fig. 293), to which another light frame, F, and a stop, S, have been secured. The furnace is heated at the point where it is to be flanged, and laid on the mould, and the flanging is then done with mallets.

Care has to be taken while doing this. If the edge of the plate were to be struck first, as shown in fig. 294, it would chiefly bend and

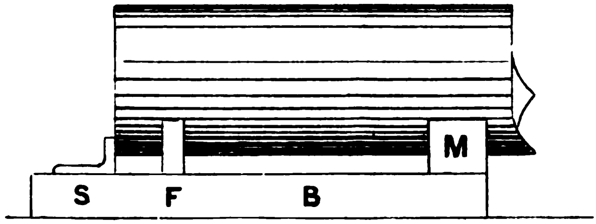
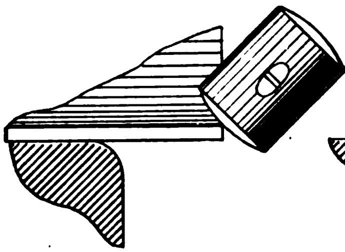


FIG. 293.



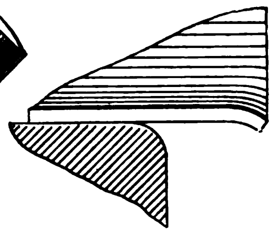
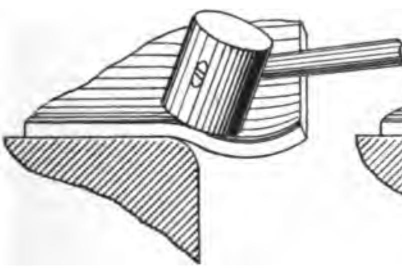


FIG. 294. FIG. 295.

stretcn the edge (see fig. 295); it is therefore usual to strike at first nearer the anvil block (fig. 296). This blow produces little effect at the



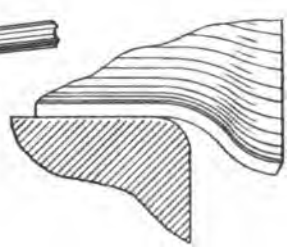
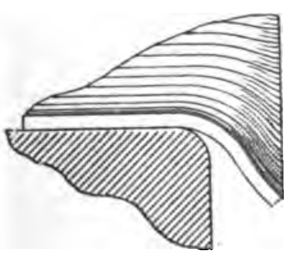


FIG. 296. FIG. 297.

edge, which therefore remains thick. Further blows produce curves, as shown in figs. 297, 298, 299.



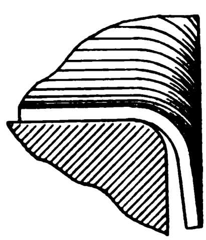


FIG. 298. FIG. 299.

By starting from the inside more pressure is exerted there, and the tendency to contracting the furnace at the saddle is reduced. It has already been pointed out that the stretching of one part of a plate sets up compression stresses in other parts, which show themselves by a swelling up (see fig. 300); but this is easily removed with the help of a facing iron, as shown in fig. 301. The effect of a hard tool is to stretch

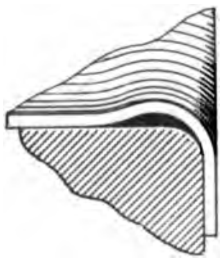


FIG. 300.

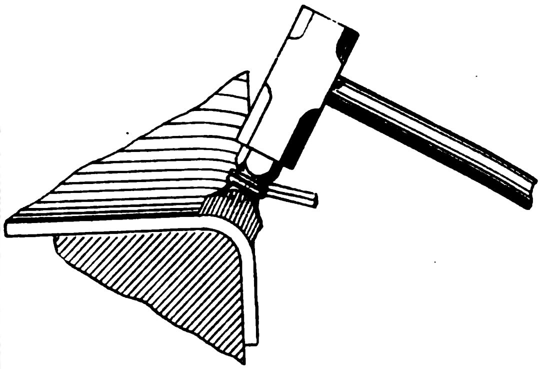


FIG. 301.

the metal locally, and that naturally increases the diameter of the furnace at that part.

This bulging was once a very noticeable feature in some patent furnaces. When allowed to remain, such faults have repeatedly been mistaken for indications of weakness or partial collapses, and much unnecessary anxiety or expense might have been saved if they had been removed at first. Very little flanging is required at the corner, for the corners help to draw this part out.

When all this work has been done the furnace will have the appearance shown in fig. 302. To finish it one of the corners is heated at

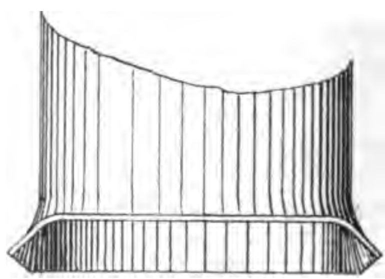


FIG. 302.

a time, and placed on an anvil mould, as shown in fig. 303. It is then hammered to the correct shape. If the preliminary flanging has not been carried out carefully, troubles may be expected not only at the corners, but also with the round parts of the furnace, which easily lose their shape.

These anvil moulds (fig. 303) may be dispensed with if the curvatures of the corners are sufficiently large, but this demands a slight deviation from the usual design ; either the furnaces have to be flanged round the tube plate, or the top corners of the saddle flanges have to be left long, and the radius of the tube-plate flange increased locally, while the combustion-chamber plate is cut away, as shown in fig. 304.

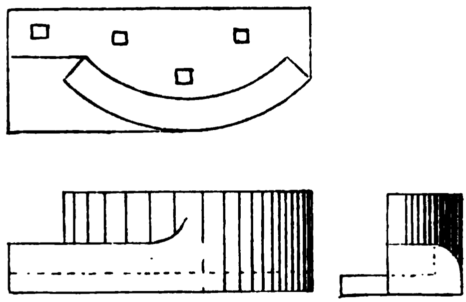


FIG. 303.

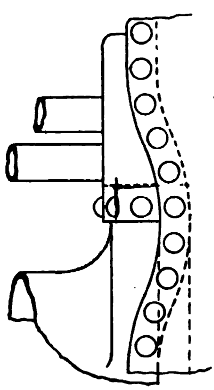


FIG. 304.

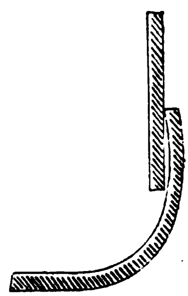


FIG. 305.

The furnaces should now be thoroughly annealed, gauged for roundness, and corrected before cooling, and, above all, the vertical part of each saddle flange should be made perfectly flat, so that it fits the flat part of the tube plate without having to be reheated. Where the flues are not supplied ready flanged by the steel works it is best to fit them and the tube plates together before annealing, and to heat and work the corners when bolted together, as is done with the end plates. If the two are fitted together after this process, it is best, but difficult, only to heat the corners of the tube plate, because it is easier to anneal it than an entire furnace. If these various precautions are not adopted it will often be found that the furnace saddle and the tube plate fit as badly as shown in fig. 305. Such work can be detected by cutting out occasional rivets.

In some works this fitting is done cold, but then the use of heaters cannot be prevented, and brittleness, due to working at a blue heat, may be the result.

In some works the furnace saddle flange is left very deep, particularly at the corners, as shown in figs. 288, 289 (p. 214). In others it is kept as small as possible (figs. 290, 291). The latter plan, particularly if the curvatures are not too sharp, is by far the best ; not only is it easier to do the flanging, reducing the number of heats from over twenty to less than ten, but, on account of the gentler treatment of the material, its liability to crack, after the boiler has been put in use, is so very much reduced that no fear need be entertained on this point.

That these troubles are not alone due to unequal expansion of

parts of the boiler, is proved by the fact that it is the corners of the central furnaces which generally crack, while those nearest the shell, where the strains are certainly most severe, do not suffer so often. But recent numerous failures cannot be said to have fixed the blame on the quality of the material, and the only alternative explanation is that the steel has been injured during flanging, either by burning it, by heating it too often or over fires giving off noxious vapours, by insufficient annealing, by manipulating the corners at a blue heat, or by overheating and straining when in use (see p. 142).

Flanging Tube Plates.—The preceding remarks make it unnecessary to add much about the flanging of tube and combustion chamber plates. As previously mentioned (p. 199), it is usual to flange the straight edges first, and to leave all the corners to the last. This practice need not be adhered to when doing the work under a press. In order to obtain the correct shapes, machine flanging must be done with suitable moulds, and then it is also of importance to shear pieces off the corners, not so much for the purpose of saving labour when trimming the edges, as to prevent the drawing out of the material. Careful measurements will show that the thickness of the metal is distributed irregularly, as shown in fig. 306. By keeping the edges of the plates hotter than the curved part this thinning action is reduced to a minimum.

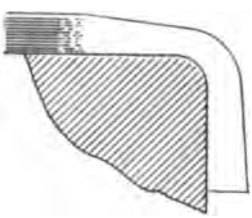


FIG. 306.

If the edges have been flanged before the corners, the flanging mould M (fig. 308) should have a good taper, so as to press the finished flanges firmly against the mould block M_1 (fig. 307); for if M is left nearly square there will be a strong tendency to draw the plate away from M_1. This is particularly the case when flanging the corners first, as shown in fig. 307; and then strong stops, S S, S S, have to be fitted. After this operation the plate should have the appearance shown by fig. 309, and the edges can then be flanged with great accuracy.

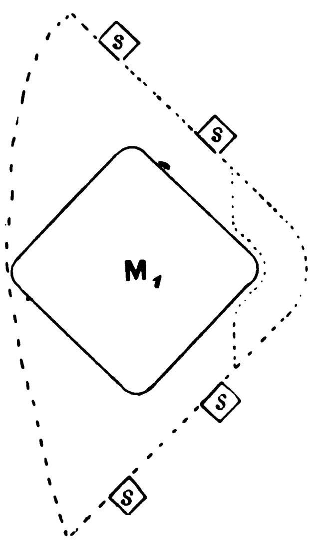


FIG. 307.

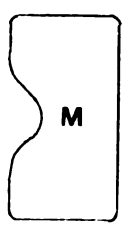


FIG. 308.

Hand Flanging. — When flanging by hand stops are sometimes used, though in some works it is deemed sufficient to make a few chalk marks on the flanging mould, and some smiths content themselves with marking the line of flange with deep centre punch marks.

The combustion chamber sides are seldom flanged. Now and then designs are met with where the back edge is turned in (fig. 310).

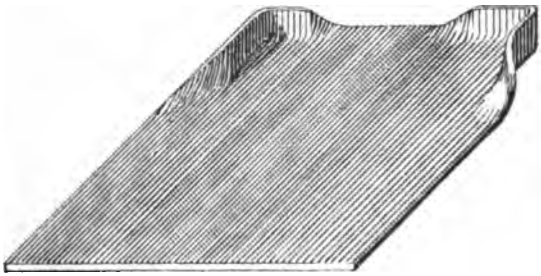


FIG. 309.

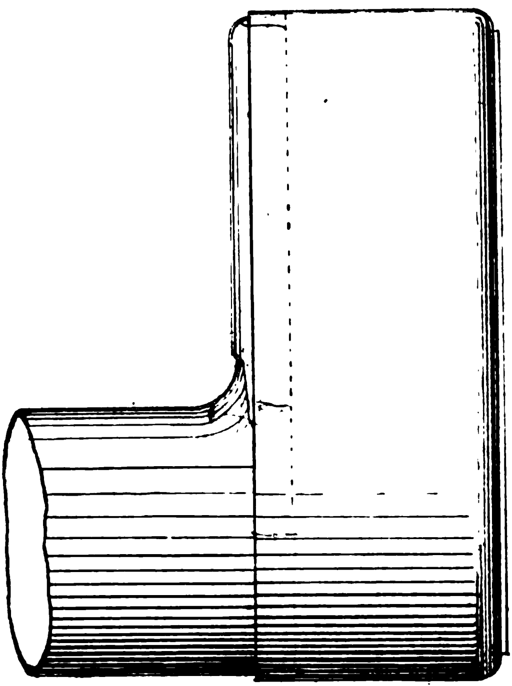


FIG. 310.

There is no advantage in this plan, and the flanging operation is a more difficult one than usual.

Another arrangement is to leave the back tube plate and combustion chamber plate flat, and to flange out the back and front edges of the top and side plates (figs. 311, 312). Unless very much rounded at the corners the flanges grow thin, or tear, and must be welded. Boilers with these combustion chambers are said to be difficult to clean, and objections have been raised against the caulking liners (fig. 312), but

they do not seem to give trouble. Some combustion chambers have rounded backs (fig. 313). These must, of course, be bent in the rolls before flanging. Some combustion chamber tops are flanged to meet the girder plates (see fig. 375, p. 234). Instead of the furnaces, the lower ends of the back tube plates may be flanged (fig. 314) like the

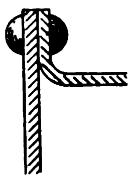


FIG. 311.

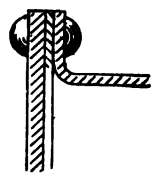


FIG. 312.

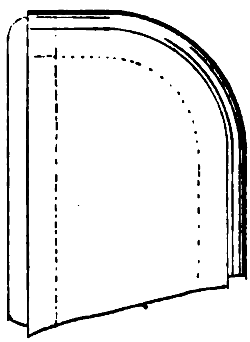


FIG. 313.

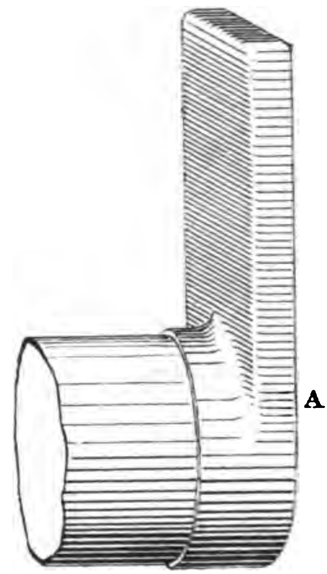


FIG. 314.

forward outer plate of the fire-box of a locomotive. It is better not to let the tube plate extend much below the centre line of the furnace, because it is next to impossible to make the two a good fit. Besides, an extra seam across the combustion chamber saves a seam round the furnace bottom. If the tube plate is made to end at A, the top part can be fitted quite close, and there will be less chance of leakage, a danger to which this arrangement is specially liable.

Reference has occasionally been made to the practice of flanging the furnace saddles round the tube plate corners—i.e. placing the back tube plate on the fire side of the furnace flange (fig. 315). It has the advantage over the ordinary style (fig. 316) of allowing both seams to be caulked efficiently; nor can steam-bubbles lodge under the landing, and it permits of the radius of the tube plate flange being made smaller than that of the furnace flange. It is asserted that the flame impinging on the caulked edge will do harm, but no trouble has ever been noticed at this point. In the one case the side corners will have to be shaped as shown in end view (figs. 288–291, p. 214) ; in the other, as in fig. 67, p. 36. In the one case the tube plate corners, in the other the furnace saddle corners, will have to be drawn out or chipped taper. As these corners have often given trouble by leaking, some works have adopted the plan of welding them.

Before concluding the remarks on this subject it is necessary to

mention an isolated practice of flanging the shell plates, instead of the boiler end plate (fig. 317). There does not seem to be any advantage, except that the upper end plates of boilers need not be annealed. The

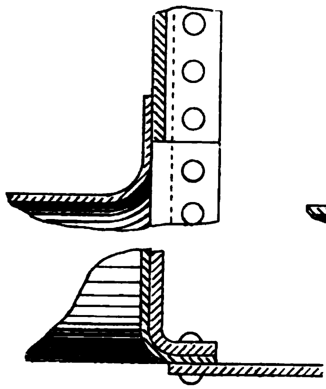


FIG. 315.

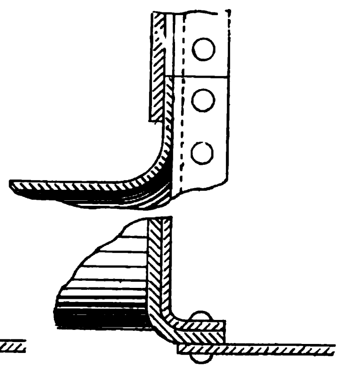


FIG. 316.

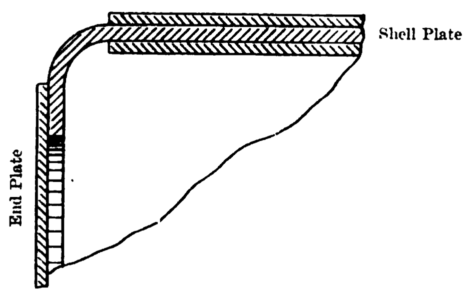


FIG. 317.

shell plate being in tension, any injury caused to these parts by the flanging operation would be doubly dangerous, as it cannot be removed by annealing.

The work is carried out as follows: The ends of the longitudinal seams are welded, the seams riveted, and the edge of the shell heated and flanged in a Tweddell's flanging press, which, to suit the requirements of the case, has to be placed on its back.

Another interesting subject is cold flanging. The results of Messrs. Easton and Anderson's experience will be found in the 'Journal

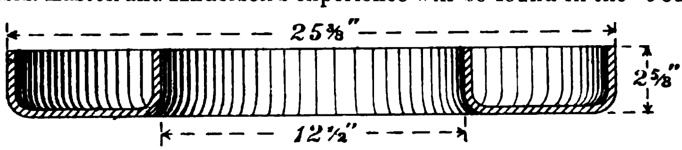


FIG. 318.

of the Iron and Steel Institute' for 1882, p. 528. They produced annular discs of the shape shown in fig. 318 with plates $\frac{5}{16}$ in. thick, but

the results cannot be said to have been satisfactory, for out of sixty plates, Landore SS quality, nine cracked their inner flanges, and one its outer flange; five were not annealed, and of these three cracked. Out of fourteen which had been twice annealed two cracked. The forty-one remaining were annealed only once, but some of them as long as forty-eight hours and in ashes. Five of these cracked, although one-half of the lot were flanged slowly, requiring about three to four minutes instead of a quarter of a minute.

No mention is made as to whether any of the plates cracked later on, though it is to be expected, for nearly all bent test pieces crack at their inner radius some time after leaving the press or hammer.

Fitting Together of Plates.—Having completed the flanging and annealing, the plates are fitted together and secured to each other by means of a few bolts. Whenever possible, the various seams are closed up cold by hammering. Troublesome parts are warmed by heaters, but, as they often reach a blue heat, it would be better to heat them properly. On this point practices differ. Some works prefer heating only one plate, bolting it to the other, and hammering it, so as to fit the cold one. This plan requires very great care in the flanging of the plate, which remains cold, as it cannot be made to alter its shape, and the danger exists that it will be made blue hot. Other works heat both plates while bolted together; then, of course, it is easy to correct any slight defects of form in both. Hardly any works re-anneal these pieces, which may account for some cracks. Less risk would be run if the corners at least were reheated.

Particular care should be taken with the flanged corners of the furnaces where they meet the back tube plate and the combustion chamber sides. Not only will injudicious treatment increase the liability of these parts to crack after the boiler has been put in use, but also, on account of the impossibility or difficulty of caulking this seam at both edges, there is a greater chance of the water forcing its way through here than through any other seam of the boiler (see fig. 413, p. 241).

Fitting Combustion Chamber Plates.—The fitting together of the various internal parts is done as follows: The tube plates are bolted to the furnaces and hammered up close, the rivet holes drilled, and the riveting carried out at once, or postponed till the other parts have been prepared. The combustion chamber back plate is then bolted to the furnace and tube plate by means of strips of iron. The sides, top and bottom plates are then successively fitted. The order in which this is done depends on the position of the seams. Representations of the various plans will be found in figs. 319-327. If the whole of the riveting is to be done by hydraulic power, the arrangement shown in fig. 325 must be adopted, the top plate being riveted before the sides are put in position; and these again are riveted before the bottom plate is secured. But this can only be riveted if the depth of the riveter is sufficiently long to reach from the furnace mouth to the back plate. If the head of the machine is a clumsy one, the flanges will have to be made sufficiently deep so that the rivets can be reached. By removing the back plate after all the side plates have been fitted it is possible to rivet at least the tube plate flange by machinery. If the riveting of all the combustion chamber seams is to

be done by hand, it is immaterial in which order the plates are put on, and preference will naturally be given to arrangements like those in figs. 319, 320. When the two plates are of equal thickness the seams can be placed higher up, as in fig. 321. Should the lengths of the plates

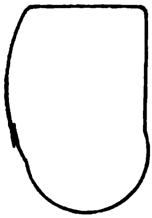


FIG. 319.

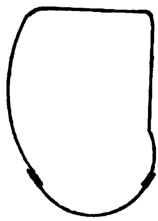


FIG. 320.

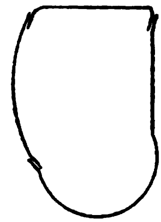


FIG. 321.

be found too great, or, what is more likely, should the back combustion chamber plate not be shaped exactly like the back tube plate and furnace, which would lead to trouble in fitting the circumferential ones in one or two lengths, it may be safer to use three plates, as in figs. 321, 322, 323, or even four plates (figs. 324, 325, 326, 327). The

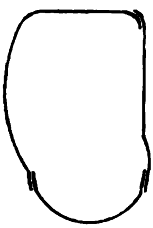


FIG. 322.

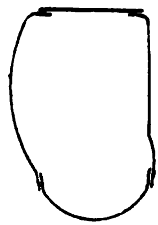


FIG. 323.

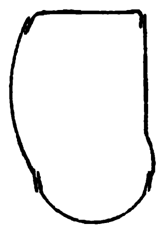


FIG. 324.

arrangements shown in figs. 319, 320, 321, 322, 324, 325 require that one or the other of the plates should be bent at two points, which, for fitting, is more troublesome than if each plate has got to be bent at

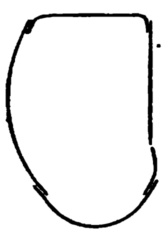


FIG. 325.

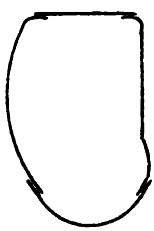


FIG. 326.

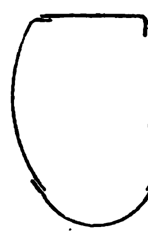


FIG. 327.

only one point, as in fig. 327. The arrangement shown in fig. 322, and on a larger scale in fig. 328, doubles the bending operations. The latter figure also shows how difficult it would be to caulk the internal edge of such a seam if placed higher up in the curve, and also that it should never be placed near a stay. (See also fig. 329.) The corners of all

these plates ought to be drawn out as shown in figs. 232 and 233 (p. 191). The various seams should also be slightly bevelled, and it is as well to do this before fitting them together.

The combustion chamber bottoms and sides are bent cold to their various curves by passing them through the bending rolls, but the corners, particularly where they have to be drawn out, are heated. It is difficult to obtain quite the correct shape at once, and the final

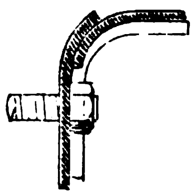


FIG. 328.

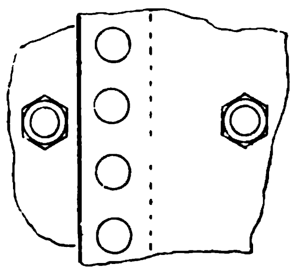


FIG. 329.

setting has to be done in place. Heaters are frequently used, but whether it is that a blue heat is never reached, or whether the work done to the plates at this temperature is not sufficiently severe, it certainly does not as yet seem to have led to any failures in the shop or subsequently.

Drilling Combustion Chamber Plates.—When the various plates have been properly fitted together, their seams are drilled in place—generally by hand, but also by machinery. In some works all the outer plates are first removed and drilled by machinery, or even punched, and then refitted, and the inner plates drilled. Sometimes the outer plates are perforated before fitting. If punched, there is always the danger that the plates may crack while bending them, or if punched after bending they are liable to warp, and part of the fitting work has to be done over again. Another plan is to drill all the flanges before fitting, to mark off the holes from the inside on the outer plates, and to drill these by machinery. In any case all holes in the flanges will have to be countersunk on the inside, for it is difficult to caulk any other heads when so near to a corner. It is also customary, but not necessary, to countersink all rivet holes on the fire side of the various lap joints; but it does not appear that projecting heads burn off. Wherever there is sufficient space for riveting, or when this is done by machinery, the rivets should be inserted from the water side of the plates.

The angle of the countersink varies from 15° to 45°. The smaller the angle, the smaller is the power of the rivet to draw the plates together while it is cooling. It vanishes altogether when the apex of the cone lies beyond the flat base of the head, or if both ends are countersunk, this limit is reached when both cones touch each other at their apexes.

Riveting Combustion Chambers.—The remarks made while discussing the riveting of shell plates, &c., apply to a certain extent to this case. The very greatest care should be taken to ensure perfect contact of the plates at the saddle and its corners, for this is almost the hottest

part of the boiler, and any air-spaces are sure, sooner or later, to lead to troubles which rapidly extend. On account of the difficulty of flanging the furnace ends, these parts are not always as flat as the tube plate to which they are connected, and carelessness in riveting easily leads to the condition of things illustrated in fig. 305, p. 217. Careless work at these seams is readily exposed by removing a rivet or two. In corners the rivets may be arranged as in fig. 330 or fig. 331, the latter of the two plans being the more generally adopted. In some

FIG. 330.

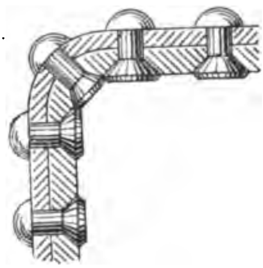


FIG. 331.

works the corner rivets are replaced by screws beaded over at either end. For remarks on caulking see p. 238.

Planing Edges of Flanges.—All the edges of the plates ought first to have been machined. With the circumferential plates this is or can be done before bending, and in that case they can be bevelled ; the flanges, unless they are trimmed with milling tools, are left with square edges. In some works they are subsequently chipped, while in others they are only chipped, and if little material has to be removed, this is done after the seams have been riveted. These remarks apply as well to the flanged end plates as to those of the internal parts. If much material has to be removed, this can be done by the shearing or the punching machine, usually before the plates are fitted ; or the superfluous parts are removed by a cross-cut chisel. The furnace saddle seams are often shaped by slotting machines or band saws.

Planing machines, lathes with very large face plates, or turntables, are also used for this work. The cutting tools or knives are made very wide, sometimes more than 12 ins., and it is then possible to deal with all sorts of irregular-shaped flanges. Of course, as there is always only one thickness of plate, only part of the width of the tool is used at a time, and the feed is a pretty heavy one. The cutting speed is about 10 ft. per minute, and the depth of cut about $\frac{1}{32}$ in. Circular saws, which cut off a solid piece of the flange, or milling cutters, have also been tried for removing the superfluous material, but are not generally used. They travel at the rate of about 2 ft. per hour. The furnace front plate flanges can be machined by the same machine (fig. 212, p. 183) which cuts the holes, but a tool with a very wide cutting edge replaces the parting tool shown there.

Fitting Internal Parts into Shell.—The furnaces and combustion chambers having been riveted together, each one has to be fitted into the furnace front plate. If all the furnaces lead into one combustion chamber, their back ends should not be riveted up without the furnace

front plate being in position, otherwise the most serious inconveniences will be encountered when trying to put the two together.

Where each combustion chamber has a separate furnace, these are fitted into the flanged front plate and their holes drilled and riveted, while the stays from one combustion chamber side to the other are screwed into place. Then, when this work has been completed the whole of the combustion chambers, furnaces, and furnace front plate are rigidly connected, and may be lifted into position in the boiler shell, to which the back plate has already been riveted.

If the front tube plate is to be placed inside of the furnace front plate, a little simple manœuvring may be necessary.

In some works the screwing together of the combustion chambers and the riveting of the furnace front seams is carried out inside the boiler shell after the furnace front plate has been riveted up. No advantage is gained, and on account of the confined space the work will be both slow and bad. Besides, if one of the furnaces should crack while its front end is being expanded, all the work just mentioned would have to be done over again.

Fitting Furnaces.—On account of the difficulty of making a furnace front plate with, say, three or four holes, slip easily on to as many furnaces, these are generally made at least $\frac{1}{4}$ in. smaller in diameter than their respective holes. Sometimes there will be even $\frac{1}{2}$ in. of difference. Then, if the flange is left square, as shown in fig. 332, the expanded furnace mouth will only bear at the front edge, as shown in fig. 333. It is, therefore, customary to leave the flange slightly conical, as shown in fig. 334. In order to get a good fit, as in fig. 335, the diameter of the front edge of the furnace will occasionally have to be increased even as much as $\frac{1}{2}$ in. It is dangerous, and in cold weather impossible, to do this expanding without heaters; and when they are used there is the further danger of overheating and of making the plates permanently brittle by hammering them when blue-hot, and cracks at these seams are not unknown.

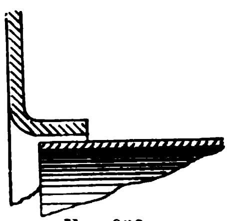


FIG. 332.

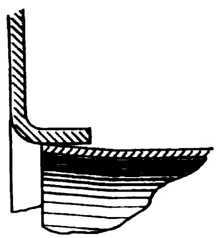


FIG. 333.

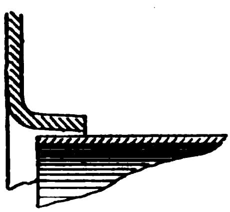


FIG. 334.

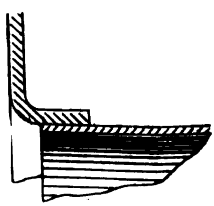


FIG. 335.

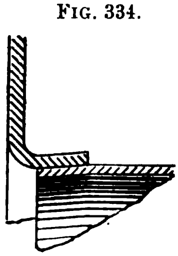


FIG. 336.

The work is carried out as follows: The furnaces are secured as centrally as possible in their respective holes, and all the holes drilled in place either by hand or by the machine (fig. 210, p. 182). Numerous bolts are then inserted, heaters applied, and the bolts screwed tighter, while the front seam is being hammered, until the plates are in perfect contact. The troubles which

these seams sometimes give have led a few engineers to make them treble-riveted, but an apparently safer and more convenient plan would be to make the flanges conical in the other direction (fig. 336), and either turn the front end of the furnace or give it a slight bevel, as shown. Instead of heaters, portable coke furnaces are sometimes used, which heat the seams almost to redness, and they can be screwed close without hammering.

The Riveting of Furnace Front Seams is not quite so simple as would at first sight appear, particularly if the water-spaces are made narrow, and there is always a very strong inclination to do this in order to gain space. With this object in view the flanges of the furnace holes are sometimes turned the other way to those of the circumferential one (fig. 244, p. 195 ; fig. 278, p. 211), or the furnace is made taper (fig. 337) or its front end contracted a few inches, or the furnace is

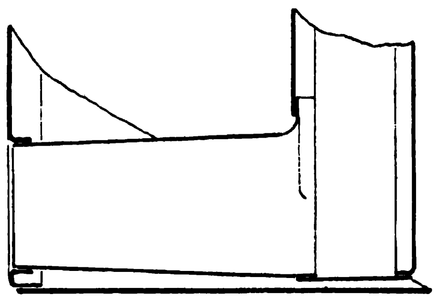


FIG. 337.

fitted at an angle. Carried to extremes this principle will be recognised in the arrangement shown in fig. 338, which would permit of the furnaces touching each other. When, in addition, the longitudinal seams both of the furnaces and the shell accidentally come together (see fig. 339), the boiler bottom is practically cut off from the upper part,

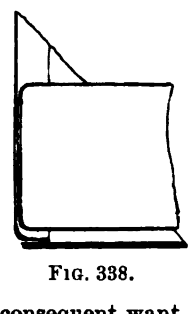


FIG. 338.

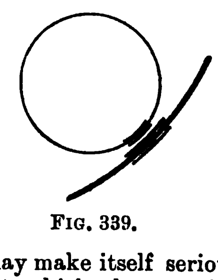


FIG. 339.

and the consequent want of circulation may make itself seriously felt. Figs. 340, 341, 342, show an arrangement which also permits of the furnaces being placed very near each other and near the shell, except the central one, because here both flanges are turned in the same direction.

In cases where two flanges are so close together that a rivet can neither be introduced nor subsequently caulked, the holes have to be

carefully drilled, tapped, and countersunk, and accurately-fitting screws, with conical heads, inserted. The ends of the screws are

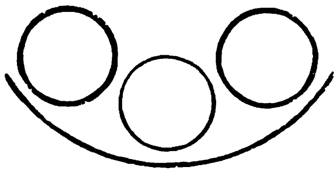


FIG. 340.

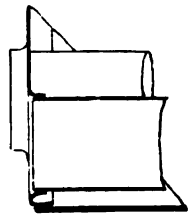


FIG. 341.

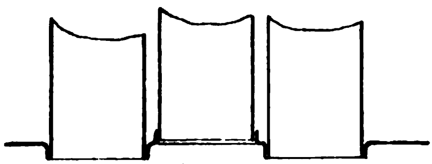


FIG. 342.

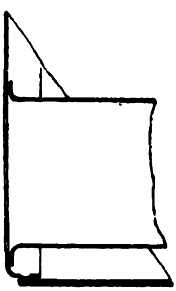


FIG. 343.

beaded over, and the heads caulked. An arrangement in which all this difficulty is overcome, and in which the front plate holes need not be flanged, is shown in fig. 343. The furnace front flanges will overlap each other. (See also fig. 41, p. 24.)

Furnace Saddles.—The various designs for securing the furnaces to the combustion chambers are shown in the following sketches. Fig. 344 is the most common, except, perhaps, when the back end seam is single-riveted all the way round, instead of being double riveted from below the line of fire bars, as shown. In through combustion chambers of double-ended boilers this seam is sometimes double riveted above and treble riveted below (see fig. 345). The increased width of the lower flange is sometimes met with in the back plate of combustion chambers (see fig. 346). In both the above cases the tube plate is on the water side of the saddle plate, and it is affirmed that unless the edge of the plate is well bevelled, and the rivet heads

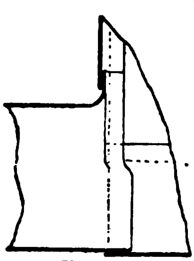


FIG. 344.

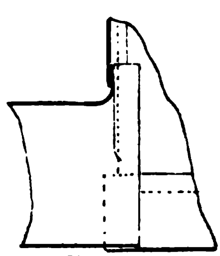


FIG. 345.

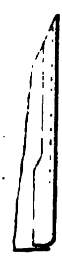


FIG. 346.

countersunk on the water side, steam-bubbles will lodge there. It is difficult to caulk this seam on the inside because of the furnace, and on the outside because of the tubes. The curvature of the tube plate side flanges is a very sharp one, in order to get the tubes as near the

edge as possible, and when the furnace saddle flange is inside of this tube plate, this curvature is often so sharp as to be more like a corner, and therefore very liable to crack (see fig. 43, p. 25). One way out of this difficulty is to increase the radius of curvature of the tube plate side flanges, where they meet the saddle corners. This part must therefore be made deeper, and it is necessary to cut away the combustion chamber side plates, as shown either in black or dotted lines in fig. 304, p. 217. Another arrangement is shown in fig. 347 (see fig. 60, p. 34). The saddle being flanged over the tube plate, it can be made with a gentler curve than in figs. 344 and 345. The saddle seam is sometimes double riveted, and believing that the large amount

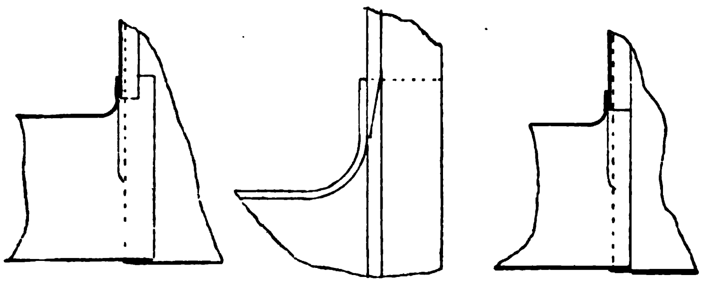


FIG. 347. FIG. 348. FIG. 349.

of metal at this point would lead to burning, some engineers plane or chip this seam (see fig. 348). When it is necessary to fit very thick plates to the combustion chamber bottoms, they are sometimes planed thin at the seam before bending, as shown in fig. 349, but no object seems to have been gained by adopting this plan.

It is objected that the saddle seams of the last three constructions expose a caulked edge to the impinging action of the flame, but in practice they answer well. It is certainly far easier to caulk both edges of this seam, and there is no projection under which steam-bubbles could find a lodgment. A similar seam is shown in fig. 350, which is adopted with ribbed furnaces, and with those fitted with several Adamson's rings. The tube plate hole is not flanged, and can be carried to the very bottom of the combustion chamber.

The arrangement shown in fig. 351 is adopted when there is little spare space between the furnace top and the tubes, but as the work of flanging is doubled, and the caulking difficult, it cannot be recommended.

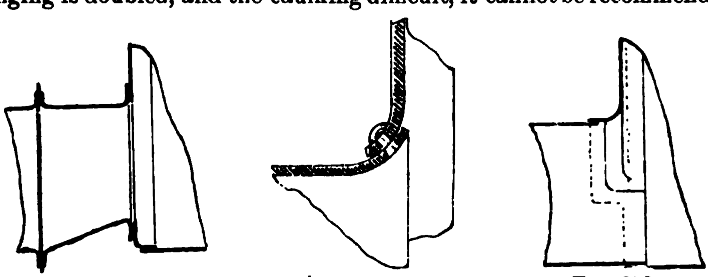


FIG. 350. FIG. 351. FIG. 352.

A somewhat more objectionable arrangement is shown in fig. 352, the tube plate being flanged to meet the furnace, and the seam being

exposed both to the radiant heat of the incandescent fuel and to the convection of the hot gases and flame.

The very greatest care has to be taken to secure metallic contact of the plates, and the flanged tube plate is sometimes bored to fit the furnace which has been turned (see fig. 353). In order to minimise the chance of leakage, and to reduce the labour of caulking, it is a good plan to remove the scale of all plates by pickling them in a 1 per cent. solution of hydrochloric acid, or to sponge the seams with sal-ammoniac. Other remarks about riveting these seams will be found on p. 32.

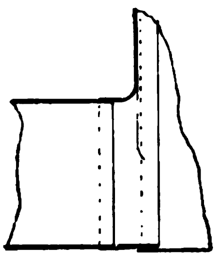


FIG. 353.

The back end seams are sometimes welded (see figs. 424, 425, p. 245), and an excellent job can be made of them, but in case of collapse they are apt to crack.

The Screwed Stays between the combustion chamber backs or sides and the back end or shell of the boiler are sometimes fitted before the circumferential seam of the front plate is riveted, sometimes afterwards. This being done by hand, all the holes will have been previously drilled or punched, and the plates annealed; they are tapped when in position, and the stays screwed and beaded over, or caulked and nutted. If the tapping and screwing of stays are done by machinery, the drilling is also done by the same machine (see p. 181). The sawing off of the ends should not be necessary, for accurate measurements of the lengths could have been taken, and it is certainly cheaper to saw off the correct lengths in a small machine than in one of these very expensive ones. It takes about 10 minutes to tap a pair of holes and to screw in the stay, and another 20 minutes to caulk and to screw up the nuts. When done by hand, the whole operation takes about one hour.

When screwed in by hand, these stays are often made with a square head (fig. 354). The labour of cutting off the ends would be saved by the use of a closed nut with a shank (figs. 355, 356). Its

FIG. 354.

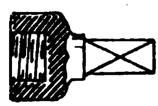


FIG. 355.

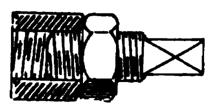


FIG. 356.

depth should not exceed the thickness of a stay nut. Stay holes are always tapped when the plates have been placed in their final position, and immediately before the stays are fitted; otherwise it would be impossible to enter them. It does not seem to matter much that the threads of the stays and of the tap are not exactly alike. This difference is due both to the tap and the stays altering their lengths respectively during hardening and during screwing.

The Pitch of Threads of the stays varies in different shops, the natural tendency being to make the pitch as fine as possible, because the effective diameter is measured from the bottom of the thread; but it takes longer to tap and screw them. There is little harm in fine threads when nuts are screwed on the ends, but a coarse and deep thread is necessary when the ends are only beaded over.

Cases in which explosions or mishaps have been traced to the use of fine threads are to be found in 'Engineering,' 1887, vol. xliii. p. 396, and 1890, vol. l. p. 85.

Should the plate bulge, as it often does when hot, its inner (water) side will leave the screw threads entirely (fig. 357), and only the outer edge will hang on to the stay by the small riveted head and by a very few threads (fig. 357); whereas with a coarse thread, as in fig. 358, it would require a serious amount of bulging before any of the threads are quite clear of each other.

The threads of the taps and stays are never quite the same, which, when fine ones are used, may cause them to strip.

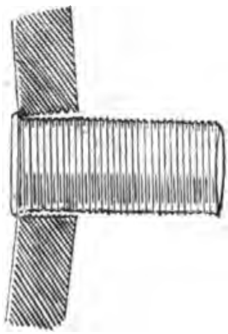


FIG. 357.

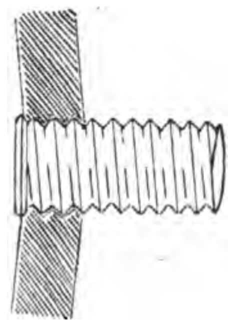


FIG. 358.

Stay Nuts ought not to be thicker, or only a very little thicker, than the plates which they have to support: not because of the danger of burning them, but for fear of stripping the thread in the plate. This is illustrated in fig. 359, where, if the full power which such a nut could stand were applied, the threads in the plate would certainly strip. The carelessness with which the holes are drilled, placing the stays at an angle, and making the nuts bear on one side only, adds to their power of doing harm. The taper washers which are then fitted often make matters worse by turning round, the thick

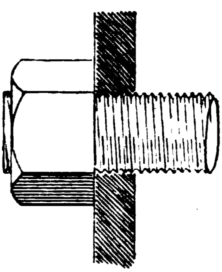


FIG. 359.

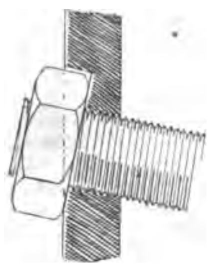


FIG. 360.

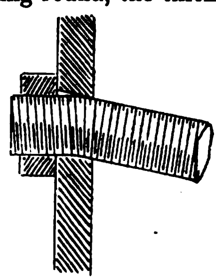


FIG. 361.

part being found where the thin part ought to be. To prevent this, the bearing side of the nut should be faced, or, better still, thick external plates should be recessed, as shown in fig. 360. Formerly the stays were bent by striking them with a hammer (fig. 361).

One often finds red lead cement, and also flat washers, under the nuts in the combustion chambers. Neither are wanted, doing more harm than good. Nobody would contend that the cement could stop any leakage, the circumference of the stay having been caulked; it can, therefore, only act as a non-conducting layer between the boiler plate and the nut, causing the latter to burn when exposed to the flame. The interposition of a washer doubles this danger. The cement is also

sure to get into the threads between the nut and the stay, where it acts as an efficient non-conductor and prevents the nut being cooled by the stay. Fig. 362 shows stay bolts which are used by some builders.

An ideally perfect stay should allow of the nut in the combustion chamber being brought into as absolute a metallic contact with both stay and plate as is possible: it will thereby be prevented from burning, and will give the most solid and efficient support to the perforated

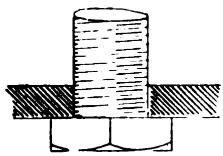


FIG. 362.

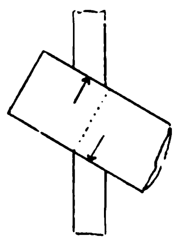


FIG. 363.

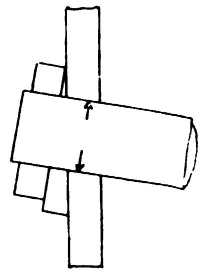


FIG. 364.

plate. Taper washers ought therefore to be dispensed with as much as possible on the combustion chamber ends of the stays, and in no case should the angle be so large as shown in fig. 363; in fact, the angle should not exceed $\frac{1}{3}\frac{t}{d}$, where t is the thickness of plate and d the effective diameter of the stay. With a $\frac{1}{2}$-in. plate and $1\frac{1}{8}$-in. stay this would give an angle of $\frac{1}{6}$, or 10° (see fig. 364).

Caulking Screwed Stays.—Screwed stays are caulked with an ordinary tool before the nuts are fitted (fig. 365). Locomotive engineers, who do not seem to like nutted stays, have used hollow bars and made them tight by drifting. Mr. Yarrow has also adopted this plan, but on account of the impossibility of getting at the outside ends, he makes the hole larger at the inner end than at the other, and uses both drifts from the fire side ('N. A.,' 1891, vol. xxxii. p. 102, plate 22). The drifts used should be fitted with threads, so that by screwing on a nut they can be drawn out again.

FIG. 365.

Girders.—These are fitted to combustion chamber tops and other flat parts in the insides of boilers, transmitting the pressure from these parts to others which are better adapted to support it. If the girders end near stays, these will have to be made of a larger sectional area, in order to support the extra load which is thrown upon them.

Most girders are made of two plates, one on either side of the stays which they have to support (figs. 366, 367, 368, 369, 370). These are connected to each other by rivets and distance pieces, and caps are fitted under the nuts of the stays.

Instead of riveting the plates together they are sometimes welded at their ends. Cast-iron feet at the ends are also used.

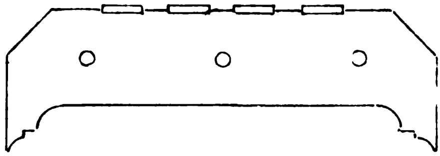


FIG. 366.

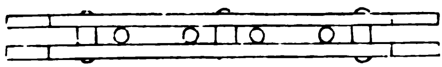


FIG. 367.

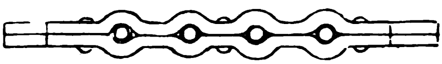
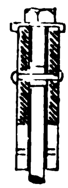


FIG. 368.



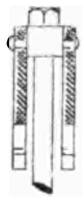


FIG. 369. FIG. 370.

Sometimes they are forged out of the solid (fig. 371) or are made of cast steel (fig. 372).

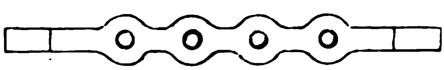


FIG. 371.

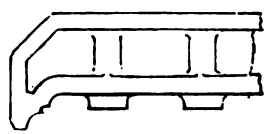


FIG. 372.

In some works the girders are arranged alternately with an odd and an even number of stays, the pitch being measured diagonally. The saving in weight, however, is too slight to balance other inconveniences.

Angle Iron and Web Girders.—Instead of girders with stays, angle irons (fig. 373), and even beams (fig. 374), are sometimes riveted to the combustion chamber tops, but this is a bad practice, as the

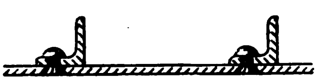


FIG. 373.

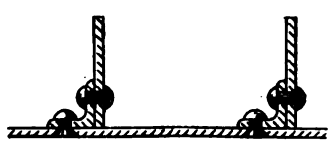


FIG. 374.

plates generally crack under the doubled parts, in the same way as they do under palm stays (see fig. 37, p. 23).

Another plan, and one which is coming into more general use, is to flange the combustion chamber plates, and rivet them to vertical webs (figs. 375 and 376). The ends of these vertical plates have to be forged with projecting feet (fig. 384, p. 235), which tuck in under the combustion chamber top plate flanges. It is not difficult either to rivet these seams or to caulk them from the fire side.

Some other shapes of girders are mentioned by D. S. Smart ('C. E.,' 1884, vol. lxxx. p. 132).

Fig. 377 shows a corrugated combustion chamber top which is occa-

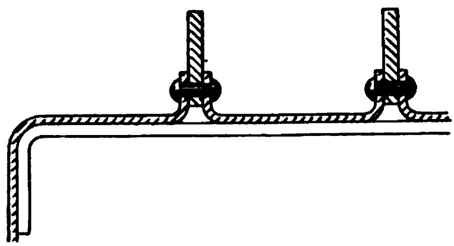


FIG. 375.

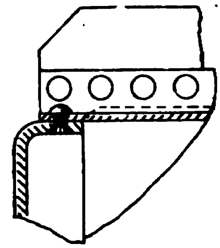


FIG. 376.

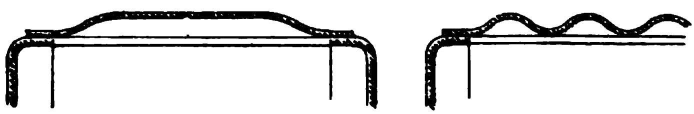


FIG. 377.

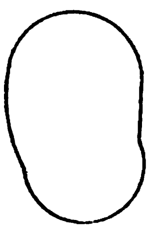


FIG. 378.

sionally fitted; its strength can be calculated with the help of a formula on p. 153. Occasionally the tops are rounded, as in fig. 378.

Suspended Girders.—Instead of allowing the girders to rest entirely on the corners of the combustion chambers, they can be suspended by stays to the boiler shells (figs. 379, 380), or their ends need not rest on the tube plate at all; but in either case the end stays should be kept far away from the flanges, in order that the expansion of the shell diameter, which amounts to about $\frac{1}{16}$ in., should not open their seams. The top ends of the suspending rods are attached to double angle irons, riveted to the shell. The top ends are flattened out as in fig. 381, so that they may

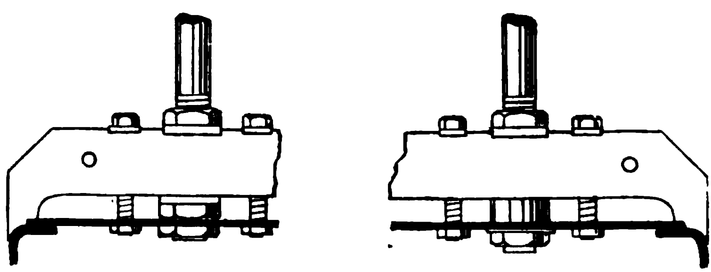


FIG. 379.

be secured by several rivets, but generally they end in an eye and a bolt, with a split key passing through their projecting end (fig. 382).

Vertical Stays.—It also happens that girders are dispensed with entirely, and every one of the combustion chamber top stays is carried

up to the shell. This arrangement is carried out as shown in fig. 382. The reason for fitting these stays to the shell is, in most cases, to relieve the tube plate of its load. But this relief may be carried

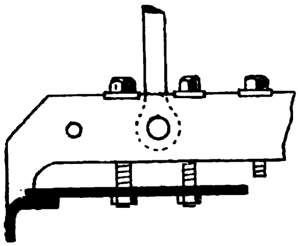


FIG. 380.

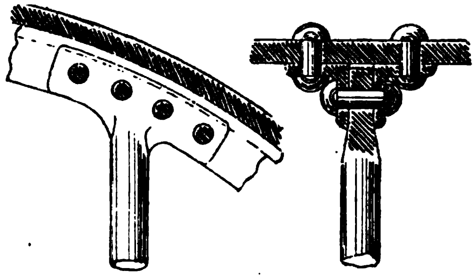


FIG. 381.

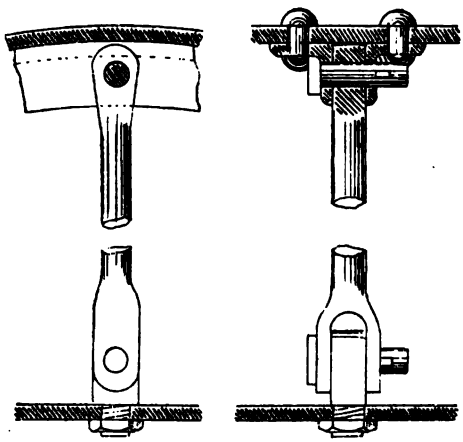


FIG. 382.

to excess, and may lead to the stretching of tube plate holes if the stays are fixed near the flanges. (See p. 29.)

Plate Stays are shown in figs. 383, 385 ; they are carried to the shell plate (figs. 383, 385) and riveted to angle irons. Fig. 384 shows

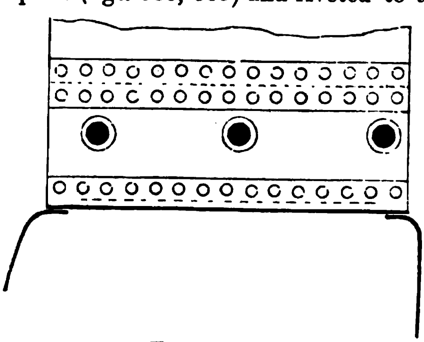


FIG. 383.

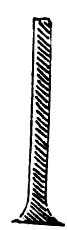


FIG. 384.

how the two ends of the vertical plate have to be stumped up so as to fill the roundings at the end.

These plates are usually fitted crossways (fig. 383). Holes are provided, through which the steam-space stays have to pass. For very

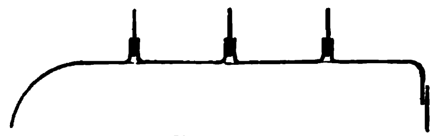


FIG. 385.

deep combustion chambers, the top has to be supported by several webs, but their number depends on the thickness of the plates.

The Steam-space Stays are sometimes tapped into the plates (fig. 386), and sometimes fitted with nuts on either side of each plate (fig.

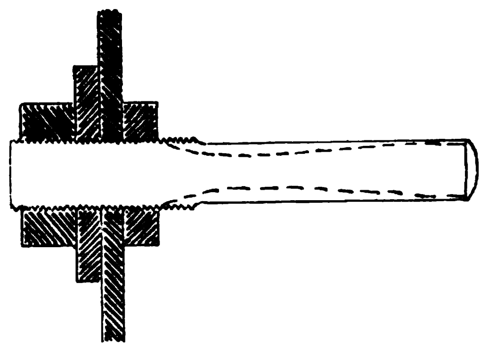


FIG. 386.

387). Both arrangements have their disadvantages. In the one case, the stays have to be swelled at one or both of their ends and threaded, then screwed into the plates, caulked both inside and out, and nutted. After being in use for some time the diameter at the front end will be

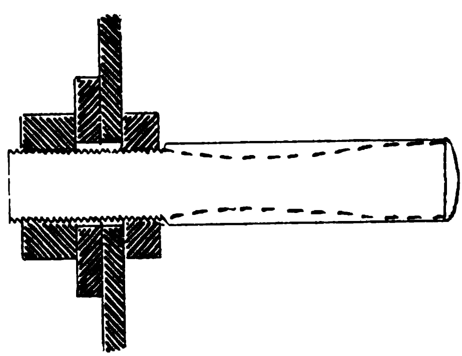


FIG. 387.

seriously reduced by corrosion, as indicated by the dotted lines (fig. 386), and the stays must be renewed. This is not the case with those stays

which have been threaded without swelling, for the exposed part of the stay may be very seriously reduced before making it weaker than the screwed part; in this case the thread at the front ends should not extend beyond the inside nut.

The trouble with these stays is that they leak, for it is only grummets and washers that can be used to prevent this. The hollow space is sometimes filled with sheet iron and caulked, but better results seem to be obtained with asbestos packing. The outside washers for these stays are very often riveted to the plate, giving it a better support. In confined spaces it may be necessary to fit the stays in two lengths. This is shown in figs. 388 and 389.

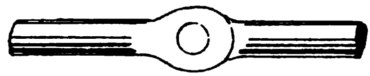


FIG. 388.

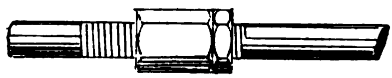


FIG. 389.

Boiler Tubes.—One of the last things to be done to a boiler is to fit the tubes. This is a simple matter: the tubes arrive cut to the right lengths and probably also annealed at their ends; they are passed through the two tube plates, and their ends are expanded. Formerly this was done by means of conical drifts, and some people still advocate their use, but the common practice is to use expanders. These consist of several—usually three—small rollers, partly projecting out of the circumference of an iron case. A taper mandril is placed in the centre and driven in while being turned round, thereby causing the rollers and their case to revolve, at the same time exerting a pressure on the tube, which expands and gets firmly bedded against the metal of the tube plate. Plain tubes can be placed in position and expanded at the rate of about six per hour.

The recent disasters with the tubes of Navy boilers, leading as they did to the appointment of a special committee of inquiry, are a sufficient proof that the mode of securing them is not a perfect one. Undoubtedly the severe conditions of Navy trials search out any defects which may exist, and when leakage has once commenced it seems impossible to stop it again until the tubes are re-expanded. By some it is affirmed that the trouble is caused by overheating the tube plate (see p. 94). Others believe that the chilling effect of inrushes of cold air, or that structural peculiarities, are to blame. Thus each tube has a slight twist, due to the expander having been worked in one direction only. On heating the tubes, they may untwist slightly. This motion will be particularly injurious if the tube plate holes are not perfectly circular. The taper shape of the tube expander which is used to force a parallel tube into a parallel hole may also be a cause.

Of the various attempted remedies, none have yet given permanently satisfactory results, and for the merchant service, where practically no troubles are experienced, no deviation from the present course would be advisable. It consists in fitting a sufficient number of stay tubes.

Stay Tubes.—These are fitted before the others, in order to be able to screw up the nuts. If the tubes are left parallel, the back end requires to be screwed sufficiently long so that the inside nut of the front end can be inserted, as shown in fig. 390. The back end is beaded over, as in fig. 393, or nutted, as in figs. 391, 394. Nowadays nearly

all stay tubes are swelled at their front end (fig. 392), and both plates are tapped together. Fig. 395 shows a section of a tool which is very

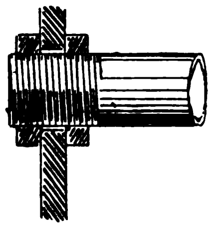


FIG. 390.

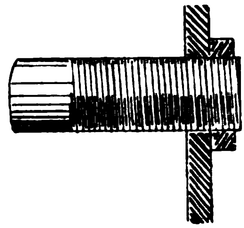


FIG. 391.

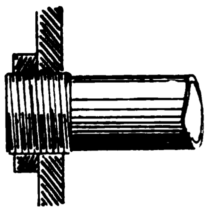


FIG. 392.

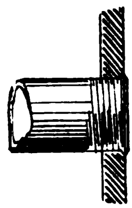


FIG. 393.

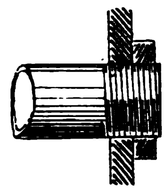


FIG. 394.

convenient for screwing or unscrewing stay tubes. Two half-round grooves, deeper than shown, are cut into the sides of a spindle which is just large enough to enter the tube, and two short lengths of steel of a lenticular section are placed into these recesses and held there by any convenient means. Their outer edges, being roughened, grip the inside of the tube whichever way they are turned. Both tube ends are expanded and also caulked, and either beaded over or fitted with nuts. The other tubes are then placed in position and expanded. Stay tubes are insisted on in the merchant service, while the locomotives and American steamers do without them, but all the ends are beaded. Exhaustive experiments on the holding power of tubes are mentioned by W. A. Shock, 1880, p. 217. As regards taper-ended tubes, experiments were carried out by Martens ('Mitt., Berlin,' 1887, vol. v. p. 65). None of these results were obtained at steaming temperatures. (See p. 30.)

FIG. 395.

The time required to tap the tube plate, fit a stay tube, and bead or nut it, is about one hour and a half. The taps are hollow, and can be adjusted on a long spindle to suit any length of tubes ; these are threaded in a lathe.

Caulking.—In order to explain what takes place during the duration of a blow, when the hammer, the caulking tool, and the plate are in contact, it will be necessary to divide the plate into layers of, say, $\frac{1}{100}$ in. in thickness, as shown in fig. 396. The velocity of the hammer and caulking tool is imparted to the first layer, and quickly transmitted to the next, and further on. The pressure which is required to transmit this velocity from the caulking tool to the first layer, and then from

one layer to another, is at first in excess of the elastic limit, and produces a permanent deformation, as shown. But this pressure reacts on the caulking tool, causing it to rebound after only a few layers have acquired the high velocity ; but, their mass being small as compared with that of the hammer, their pressure on the further layers is not sufficiently great to flatten and spread them out. The permanent effect will, therefore, be that which is illustrated in fig. 396. A little reflection will show that, the lighter the hammer, the sooner it rebounds, and the fewer the layers acted upon, and the lighter the blow,

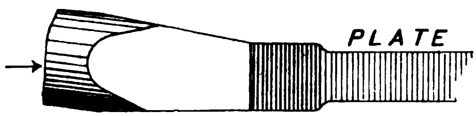


FIG. 396.

the slower the imparted velocity, and the smaller the deformations. Thus, if twenty-five layers equal to $\frac{1}{4}$ in. are deformed by the blow of a 7-lb. hammer on a 2-lb. caulking tool, only half that number, equal to $\frac{1}{8}$ in., would be spread out when using a hammer weighing $3\frac{1}{2}$ lbs. and a 1-lb. tool. By increasing the velocity, the force between the caulking tool and plate would be increased, and the swelling would be greater, but the distance to which this swelling extends would not be altered—at least, not materially. In an exaggerated form the effects would, therefore, be as shown in fig. 397. For this reason heavy hammers are first used, and then light ones.

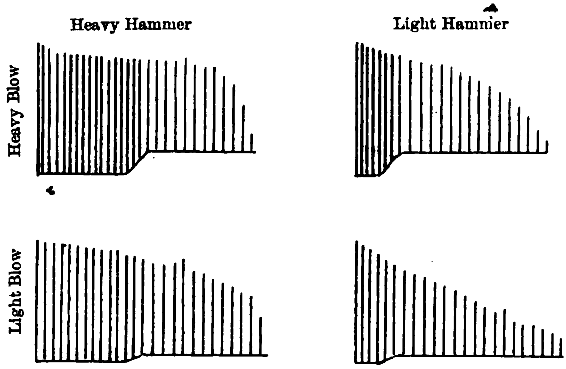


FIG. 397.

The next thing to be considered is the shape of the tool and the shape of the edge of the plate. In fig. 398 the edge of the plate is square, and the effect of the caulking tool is seen under it (fig. 399). The metal has simply been swelled up.

In fig. 400 the edge of the plate is bevelled, and the caulking tool is placed firmly against it. The result of the blow is seen below (fig. 401). Not only is the edge of the plate swelled up, as in the previous case, but the lower plate is scraped up, forming a small ridge : and, thirdly, the blow being directed downward, both plates are depressed ; but as

there is more spring in the outer plate (lap), it will not suffer as much permanent deflection as the lower one, and the result will be that

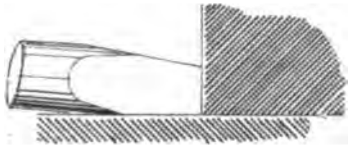


FIG. 398.

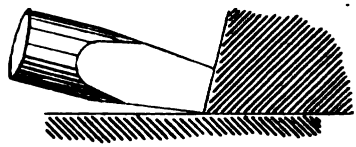


FIG. 400.

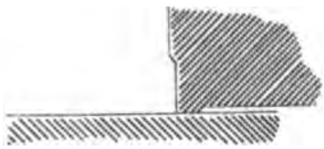


FIG. 399.

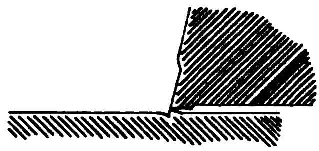


FIG. 401.

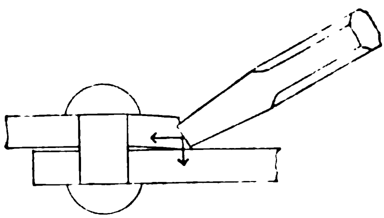


FIG. 402.

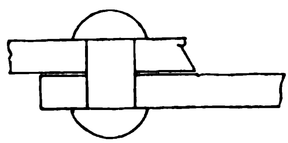


FIG. 403.

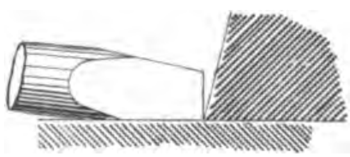


FIG. 405.

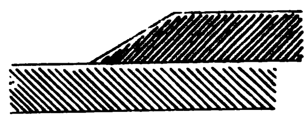


FIG. 404.

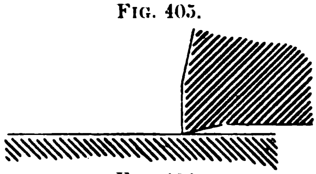


FIG. 406.

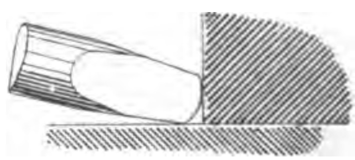


FIG. 407.

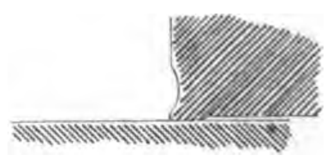


FIG. 408.

the edge remains slightly, but permanently, open. This view is further illustrated in figs. 402, 403. An exaggerated seam of this sort is shown in fig. 404, and it is quite clear that no amount of hammering on the slanting surface would caulk the joint effectively. The maximum angle met with in practice is 1 : 3.

A better result would be obtained by placing the tool as shown in fig. 405, but, on account of the thinness of the edge, the caulking

would not be very deep (fig. 406). Another method is shown in figs. 407 and 408, and strongly recommended by locomotive engineers. The effect of a blow in this case is almost the very reverse of the previous one; for while the edge of the metal is being swelled, the plate under it is being struck down, and, as the hammer rebounds, the lower plate springs back and presses firmly against the caulked edge (see figs. 409, 410). A tool of this shape will cause the swelling to extend farther into the plate, and thereby give a larger bearing surface.

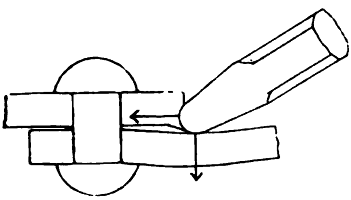


FIG. 409.

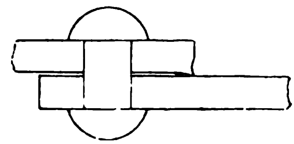


FIG. 410.

Care should be taken not to make this tool too small, otherwise it will act like a wedge, and press up the outer surface (fig. 411). Special tools have to be used for inside corners and for some rivets; they are generally shaped as shown in fig. 412. It is painful to handle them, and their work is never very satisfactory. The jar of the blow on all caulking tools is greatly reduced, if their ends are grooved, by striking them on a coarse file in a red-hot condition.

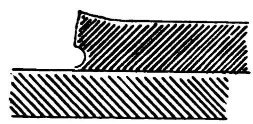


FIG. 411.

The time required for caulking is about fifteen to eighteen minutes per foot, and five minutes for one rivet head.

There is apparently no need for caulking the inside edges of seams, for they could never be relied upon for water-tightness; but when the plates are not close, as shown in fig. 413, the inside edge should at

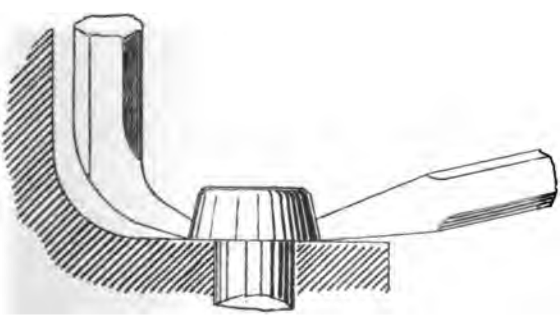


FIG. 412.

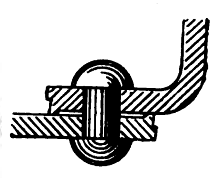


FIG. 413.

least be fullered to prevent a rocking of the plates and an occasional opening of the outer edge or loosening of the rivets.

Pickling.—Riveted seams are said to be tighter if all black oxide scale is first removed from their surfaces, and the Admiralty practice

with regard to tube ends is to grind them bright, so that they shall be in metallic contact with the tube plate. The pickling fluid which is used for removing the black scale consists of 1 per cent. of hydrochloric acid in water. Sulphuric acid, more than any other, has the effect of making steel brittle, particularly the hard qualities (see p. 113).

Welding Operations.—Reference has already been made to the fact that many of the seams of the internal parts can be welded, thereby saving the labour of flanging, drilling, riveting, and caulking. At one time efforts were made to weld iron shell plates, but the results were not encouraging, many seams having subsequently to be covered with straps. The introduction of steel, and the difficulty of welding it, stopped all progress. For a time it was only the furnaces which were welded; for, as these are not subjected to circumferential tension strains, the Board of Trade and Lloyd's Register raised no objection. It is, however, well known that even now great difficulty is experienced in keeping the welded seams of some patent flues intact during the process of manufacture.

No doubt can be felt that better results are now obtained than formerly: they are doubtless due to improvements in the production of the milder qualities of steel (20 to 25 tons), which can be made almost absolutely free from sulphur and phosphorus, two of the most injurious impurities. The influence of various chemicals does not seem to be accurately known, but what information could be collected on this subject will be found on p. 105.

The most reliable test for ascertaining whether steel is weldable is to cut off a strip about 18 ins. long, heat its centre, bend it and weld it, bend back the two ends (fig. 414), and then pull the sample asunder in the testing machine. The welded surfaces will probably be smooth and bright, except at the edges, where small patches of metal have left one side and stuck to the other. The larger these patches, and the

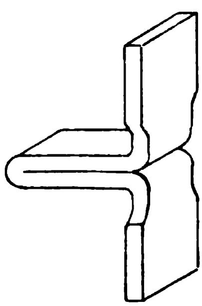


FIG. 414

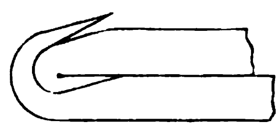


FIG. 415.

greater their number, the more weldable is the steel. This test is not applicable to iron, as this metal tears through at the corners. To test it welded joints are re-heated and bent; if good, the weld should not open (see fig. 415). The ordinary tensile test is of little use when applied to welded samples, for, provided the joint is sufficiently taper, the strength will appear satisfactory. A far better test is to trepan small rings out of a welded plate, tap them, and tear them asunder by means

of two screw plugs (fig. 416). Very serious defects have occasionally been exposed by this means.

Having obtained a good material, several precautions have to be taken to insure a good weld. The heating should be carried to the right point. This knowledge can only be gained by practice. Steel, unlike iron, should not be heated till sparks make their appearance.

Both sides of the plate should be exposed to the fire. If this is impossible the joint must be left wide open, to allow the flame to pass to the other side, where a cap of firebrick is placed. Having completed a short length of weld, and while heating the next, care should be taken to let the escaping flame pass over the recently welded part so as to keep it hot, otherwise the resultant irregular contraction will produce cracks.

The anvil on which the welding takes place should be as solid as possible. Any appreciable amount of spring affects the quality of the seam.

Fluxes are used for iron but seldom for steel. The edges of the plates, whether of steel or iron, are slightly tapered, as shown (fig. 417),

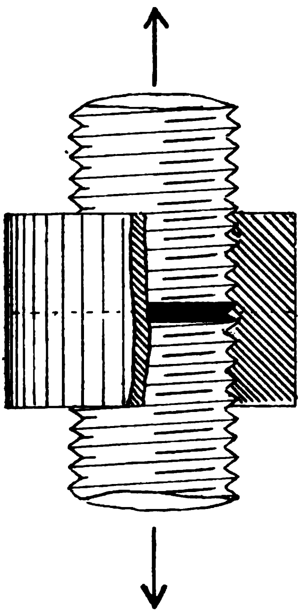


FIG. 416.

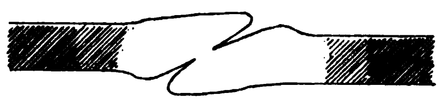


FIG. 417.

FIG. 418.

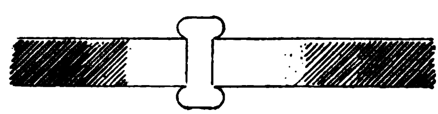


FIG. 419.

care being taken to keep the surfaces convex, so that the centres touch first and the slag thereby gets driven out. Or the edges are formed into V's (fig. 418). They are first pressed firmly together, and then hammered. A very convenient method of welding steel plates together is to insert a separately-heated good weldable iron bar into the seam. Where practicable this piece is shaped like a double-headed rail (fig. 419).

Fires for Welding.—The heating of the plates is usually done over

coke fires, using a blast. Much time is saved if these furnaces are filled and replenished with red-hot coke taken out of an adjoining furnace. The air blast for welding ought not to pass through more than about 18 ins. of coke, otherwise the combustion is not perfect, and the flame not hot enough. (See W. van Floten, 'Stahl und Eisen,' 1893, p. 26.)

Fig. 420 shows a movable furnace delivering its flame in a horizontal direction. It can only be used with plates placed vertically, and then it is usual to place one furnace on each side.

Longitudinal seams in furnaces are very conveniently heated as shown in fig. 421. As the flame only strikes one side of the plate, it is necessary to prevent radiation by the little firebrick cover C. It is

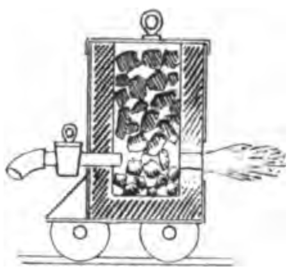


FIG. 420.

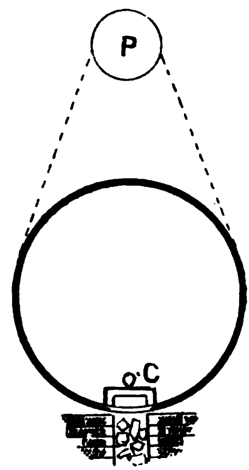


FIG. 421.

also necessary to keep the seam wide open at this point, by bending back its edges, so that the flame can pass through it; otherwise only one side of the plate will weld. When ready, the furnace plate is lifted and turned round by means of the pulley P. This can be dispensed with by placing the furnace inside of the flue, as shown in fig. 422. In

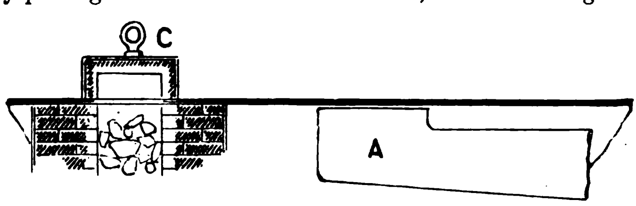


FIG. 422.

this case, when the seam has been sufficiently heated it is placed over the projecting anvil A, and welded.

Where gas is used for heating purposes, the arrangement shown in fig. 423 is a very convenient one, the ignited mixture of gas and air passing down the tube T. A is the anvil, and H the steam hammer.

It is dangerous to use fans for supplying the air to the gas, as their action is not so reliable as that of any of the positive blowers, and causes explosions.

When the tube plate and furnace are to be welded together,

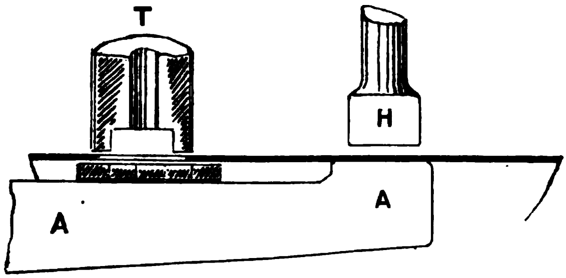


FIG. 423.

this is usually done at the corner (see fig 424). No flanging is then necessary.

But another plan may be mentioned according to which the tube plate is first flanged so as to form a saddle, and this part is then welded to the furnace (fig. 425). This method is particularly convenient for

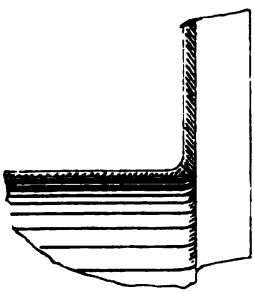


FIG. 424.

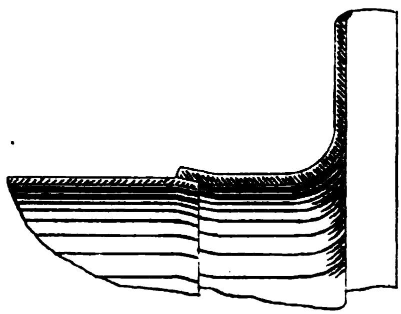


FIG. 425.

use with patent flues, which, with one exception, are weakest near the combustion chamber end. With the above plan the thick metal of the tube plate supports the flue.

Although not carried out to any great extent, all the seams of the furnaces and combustion chambers can be welded, and the finished article will then have the appearance shown in fig. 426.

Electric welding is said to give excellent results, but has not yet been used for boiler work. Welded seams are tested with oil to see whether they are perfect.

Hydraulic Test.—Although not forming part of the construction, the final hydraulic test is the concluding operation before the boiler is put into the ship. It is usual to have a preliminary test the day before, when any defective caulking can be made good, but with good work this should not be necessary. The generally adopted plan is to raise the pressure step by step, and at once caulk any defect which

shows itself. If the full pressure is put on at once, the leakages may be so excessive that they cannot all be put right. During this preliminary test the stay nuts are not screwed on, so that the stays may if necessary be recaulked.

Much has been said and written against testing boilers to double the working pressure, but, in spite of assertions to the contrary, riveted seams, and even welds, which were found to be perfectly tight with the cold test, commenced to leak at half that pressure when hot. This is

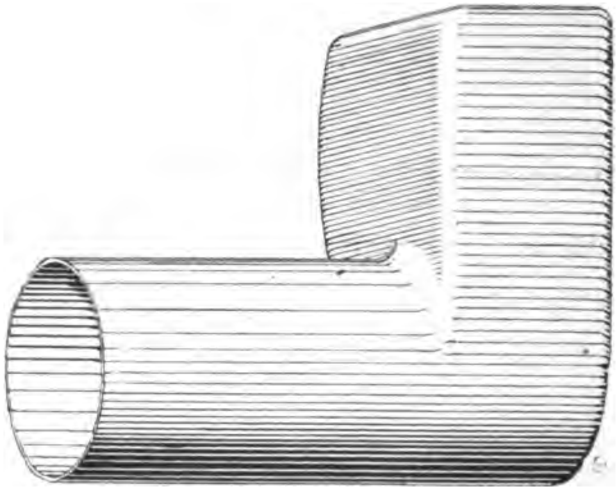


FIG. 426.

probably due to the difference in the conditions of testing, and not, as is often stated, to the previous excessive proof stress.

It is also argued that an hydraulic test not exceeding the working pressure will detect defective material and flaws. This is not borne out by experience (see p. 171).

In one sense, the hydraulic test can therefore be looked upon as a guarantee of good material, although its chief object is to detect bad workmanship. This will show itself by leakages.

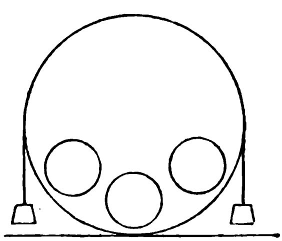


FIG. 427.

The amount of water which has to be pressed into a boiler amounts to about $\frac{1}{3}$ per cent. of its gross volume, unless all the air has not been removed, or unless the boiler is weak, in which case more water has to be pumped into it.

Boiler Deformations.—Observations as to the deformations are also made, but it cannot be said that they have been of much assistance in detecting local weaknesses; but they undoubtedly offer a means of studying the actual stresses in boilers, and a few remarks on the subject will not be out of place. That they are very much larger than most engineers expected was shown

during the discussion on J. T. Milton's paper ('N. A.,' 1893) on this subject (see p. 134, &c.).

The stretching of the boiler circumference can be roughly measured by coiling a wire $1\frac{1}{2}$ times round it (fig. 427) and weighting the two ends, and then marking the two wires at the top of the boiler before and during the test. The stretch ought to be about $\frac{1}{1000}$ of the circumference, or, say, $\frac{3}{8}$ in. for an 11-ft. boiler, and $\frac{1}{2}$ in. for a 13-ft. boiler. It will be found that the reading is less, and also, on relieving the pressure, that the original marks do not coincide. This is not permanent set, but is due to the friction between the wire and the shell. More accurate results can be obtained if one end of the wire is bolted to the boiler and small rocking frames interposed between the shell and the wire at other points (see fig. 428). The wire is secured by means of a small bolt to the rivet-head A, and led from one rocking frame to another till it reaches R, and the weight W is then hung to the loose end, while a scale is secured partly to the bolt-head A and partly to the wire at V. The rocking frames (fig. 429) are made of thin steel plates, sharpened at the bottom. The wire passes over the notch y. By fixing several wires round one

FIG. 428.

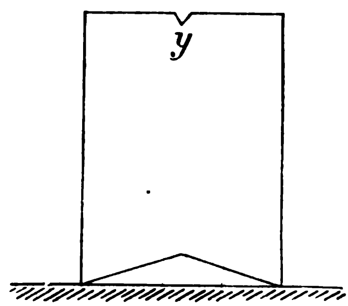


FIG. 429.

boiler the straining of the various parts of the shell and the influence of the end plates and seams can easily be ascertained (see p. 149).

Most of the other parts of the boiler can best be measured by rods or battens having a micrometer screw or other contrivance attached to their ends. Fig. 430 shows a convenient instrument for this purpose; S is one of a large number of small sleeves, to which ordinary wood screws have been brazed, and by which means they are secured to the wooden rod or batten; R R is a steel rod which easily fits the sleeve S, and is accurately graduated, preferably in millimètres; V is another sleeve, slit open on one side, and containing a vernier. The two should be a sliding fit. Measurements of the various points

of a boiler before and after testing are taken by consecutively inserting the rod R, R, with its vernier into all the sleeves S and noting the readings (see p. 133).

For the measurement of furnace deflections, ordinary trammels are

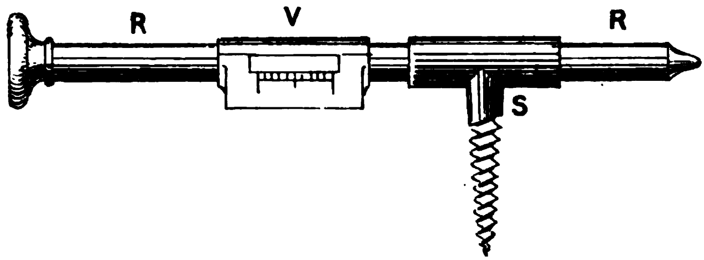


FIG. 430.

more convenient, as, by their means, it is easier to detect the major and minor axes of deformation.

For instance, if measurements are taken at the axes A A and B B (fig. 431) with a deformation as shown by the dotted lines, then the final readings would differ but little from the original ones. However, if trammels are used, they will show that the diameters had changed their angle, and the trammel resting at A would touch *e* instead of *a*, while the one resting at B would touch *f* instead of *d*. This would lead to further measurements being taken at *jh* and *ik*, and would

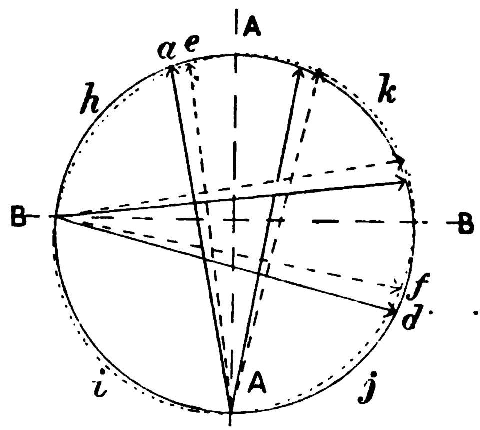


FIG. 431.

show that a serious deformation was taking place there. Should one of the diameters be so much reduced that the trammel cannot be got into position, its points of touching must be marked *along* the axis of the flue.

In order to reduce these measurements to absolute readings, employ the following formula: Δ_1 and Δ_2 are the difference between the

diameter of the furnace and the length, D, of the trammel ; x and y are shown in fig. 432.

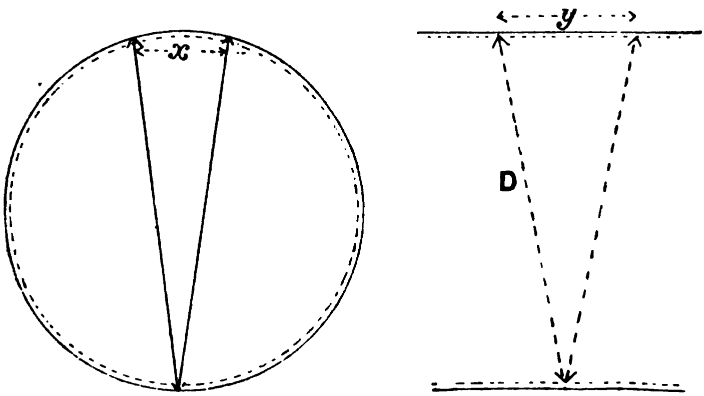


FIG. 432.

$$\Delta_1 = \frac{y^2}{8D}. \qquad \Delta_2 = \frac{x^2}{8D}.$$

If the original reading was x and the subsequent one y, then the deformation of the furnace was $\Delta_1 + \Delta_2 = \frac{x^2 + y^2}{8D}$.

Let $D = 40$ ins., $x = 4$ ins., $y = 2$ ins., then

$$\Delta_1 + \Delta_2 = \frac{16 + 4}{8 \times 40} = \tfrac{1}{16} \text{ in.}$$

If the first measurement of x was 6 ins., and the second one 4 ins., then the deformation was also $\frac{1}{16}$ in., viz. $\frac{36 - 16}{8 \times 40}$.

Having tested the boiler, it ought to be examined internally, after which it is usual to cement the lower seams of the shell, to cut the various openings for steam pipes, &c., and to fit the boiler on board.

On account of the numerous recent cracks in furnace saddles, it has been suggested that these parts should be severely hammered after the test.

CHAPTER VIII.

DESIGN.

The Proportions of Heating and Grate Surfaces and sectional areas of tubes, funnel, and other parts of boilers vary considerably according to the experiences of the manufacturers or shipowners, and it would be rash to attempt to harmonise these divergent views; all that can be attempted is to reduce the problem to the simplest elements, and these, it is hoped, will indicate how to make comparisons and draw conclusions from authentic records. The two conflicting factors are economy on the one hand, and maximum performance for a given weight of boiler on the other. In the one case large heating surfaces are necessary, in the other a high temperature of the escaping products is essential.

Assuming the case of a boiler supplied with feed water of 100° F. and worked at a pressure of 150 lbs. (water temperature 315° F.), and assuming that the fuel is capable of evaporating 15 lbs. of water at and from 212° F., equal to 12·3 lbs. under the above conditions, and assuming that 15 lbs. of steam are required for each I.H.P., then the following results may be expected:

Pounds of water evaporated per square foot of heating surface per hour	2 to 3	5	9
The evaporative efficiency of the boiler will be approximately	70 %	65 %	60 %
The uptake temperatures will be about	500°	700°	900°
Weight of water in lbs. evaporated per lb. of fuel	8·6	8·0	7·4
Weight of fuel per hour per I.H.P.	1·75	1·88	2·03
Square feet of heating surface per I.H.P.	5 to 7½	3	1⅝

With the help of the above-mentioned funnel temperatures it is possible to estimate the force of its draught when the height is known.

Funnel Draught Measured in Inches, Water.

Mean Temperature of Waste Gases	Mean Height of Funnel above Fire Bars				
	20 ft.	40 ft.	60 ft.	80 ft.	100 ft.
400° F.	·14	·28	·42	·56	·70
600° F.	·17	·34	·51	·68	·85
800° F.	·19	·38	·57	·76	·95
1,000° F.	·20	·41	·61	·82	1·02

Any desired pressure or suction can of course be obtained by mechanical means (forced draught). If the specific resistance due to fuel and other obstructions were known, it would be easy to estimate Q, the weight of air which passes through 1 square foot of grate per hour: $Q = 730\sqrt{\frac{h}{1+r}}$, where h is the draught pressure in inches of water, and r the specific resistance to the air passage; but as this latter value can only be guessed at (see p. 77), and as the weight of coal consumed is not strictly proportional to the weight of air which passes through the grate, the matter is simplified by comparing the coal consumption and funnel draught direct.

Approximate Coal Consumption in lbs. per Square Foot of Grate.

Draught pressure, inches (water)		$\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	1	2	3	4
Pounds of coal burnt per hour per sq. ft. of grate	from	15	20	25	30	40	50	60
	to	20	25	30	40	50	75	80

The following case will illustrate the use to which these tables can be put. The relations between the various surfaces are to be determined for a boiler whose funnel height is 80 feet, and which is to give economical results. The uptake temperature should therefore not exceed 500° F., and the draught suction will be about ·6 in.; the consumption per square foot of grate will be about 20 lbs. per hour; the indicated horse-power will be about 11 per square foot of grate, and the heating surface will have to be about 55 times as large as the grate area.

If economy is of secondary consideration, then the uptake temperature will have to be about 1,000°, the draught will be ·85 in., the consumption 30 lbs. per square foot of grate, and the horse-power about 14, which is not much more than in the above case; but the heating surface will be very much less, viz. 23 times as large as the grate.

Had the funnel been shorter, the consumption per square foot of grate would have been less, and the ratio of the heating surface would also have been reduced, whereas with forced draught it would have to be increased.

The absence of any reliable data on the relationship between funnel height, grate area, and heating surface makes it difficult to check the above values by actual performances, and the above tables and calculations should, therefore, only be looked upon as an indication how deductions could be drawn from really reliable information. The necessity for doing this has occasionally made itself felt in boilers which were to be exceptionally economical. In one case the lowness of the funnel and its temperature reduced the boiler performance so seriously that it was necessary to reduce the heating surface by blocking up a large number of tubes; and it is evident that if these boilers had originally been properly designed, they could have been made of very much smaller dimensions.

Funnel Dimensions.—A natural desire to reduce both the height and diameter of a funnel, so as to offer little resistance to the speed of the ship, will, if carried too far, reduce the draught, and with it the boiler performance. Many people hold the view that a short funnel with a large diameter is as efficient as a tall one with a reduced sectional

area; but this can only be true in cases where the resistance to the motion of the products of combustion is greatest in the funnel. In well-designed ones this is far from being the case, and then, within certain limits, the draught is not affected by the diameter. The one limit is determined by the velocity of the waste gases, which should not exceed 25 ft. per second under natural draught. The other limit is more difficult to fix, but it is quite certain that if a funnel is made too large in section, cold air will rush down from above and interfere with the up current. This happens with those funnels which occasionally draw well, and at other times badly. It is well known that factory chimneys parallel outside and conical inside, which were the fashion some years ago, were serious offenders in this respect, from which it is reasonable to conclude that not only the size and proportions, but also the shapes of funnels, affect the results.

Dimensions of Grates and Furnaces.—As a general rule, and for ordinary work, the stoking of furnaces whose bars are longer than 5 ft. cannot be done economically. When their lengths exceed 6 ft. most of the extra coal burnt is simply wasted. This is particularly the case with forced draught, and some engineers advocate that under such conditions the grate should not be longer than 4 ft.

Stoking is seriously interfered with if the furnace diameters are small, and probably the cooling influence of the plates close to the fire retards combustion, whereby unconsumed gases are permitted to escape. So that, unless grates are very short, furnaces should not be made less than 3 ft. diameter, and where flaming coals are used they should be still larger.

Builders prefer small furnaces, because with them more grate surface and more heating surface can be got into a boiler of a given diameter; but steam users should see that this does not lead to their being supplied with an inferior boiler.

Tubes.—The furnace and tube lengths are practically identical, and vary from 4 ft. for short double-ended boilers to 9 ft. for single-ended boilers with forced draught. In the latter case the tube diameters are about $2\frac{1}{2}$ ins., while for natural draught, where a small internal sectional area would offer too much resistance, the length is usually about 24 diameters.

Tube Surfaces, and Space Occupied.—The following table gives some of the dimensions of boiler tubes. The thickness of the metal is assumed to be ·15 in, or about No. 9 wire gauge. The external and the internal heating surface are given per foot of length. The internal sectional area of a single tube is also added. The last two lines of the table contain the amount of end space required by one tube when it is surrounded either by 1 in. or $1\frac{1}{4}$ in. of water space (see dotted lines, fig. 433).

External diameter, inches	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{3}{4}$	3	$3\frac{1}{4}$	$3\frac{1}{2}$	$3\frac{3}{4}$
Sq.ft. of external surface per ft.	·59	·65	·72	·79	·85	·92	·99
„ internal „ „	·51	·57	·64	·71	·77	·84	·91
„ „ sectional area	·021	·029	·033	·040	·048	·056	·065
Square inches of end space per tube.							
Water spaces = 1 in.	10·5	12·2	14·0	16·0	18·1	20·2	22·6
„ = $1\frac{1}{4}$ in.	12·2	14·0	16·0	18·1	20·2	22·6	25·0

In one and the same boiler the number of tubes for each furnace sometimes differs considerably. This is a bad design, for there are either too few for the one or too many for the other. The tube surface amounts to from 75 to 85 per cent. of the total heating surface.

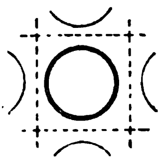


FIG. 433.

Boiler Diameters. — Having decided how many square feet of tube heating surface the boiler should contain, what diameter and length the tubes should be, and what widths of water space are to be left between the tubes, the above table will enable one to estimate how many square inches of tube plate area will be taken up by the tubes. These areas are represented by A and B (fig. 434), and the various letters indicate the other water and steam spaces, viz. :

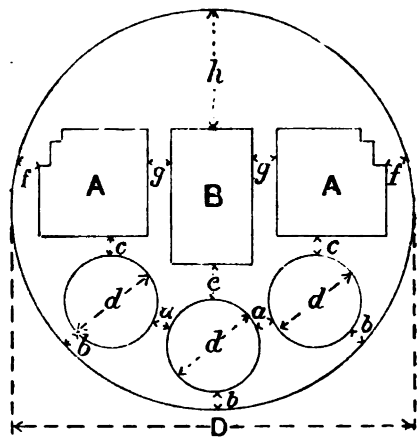


FIG. 434.

- a is the distance between two furnace diameters.
- b is the distance between the furnace and the shell.
- c is the distance from the wing furnace to the lower row of tubes.
- d is the furnace diameter.
- e is the distance from the centre furnaces to the lower row of tubes.
- f is the distance from the shell to the wing tubes.
- h is the same dimension measured in the steam space.
- g is the distance between the nests of tubes.

All these dimensions are to be taken, not from centre to centre, nor from the circumferences, but from the squares surrounding the tubes, as shown in fig. 435.

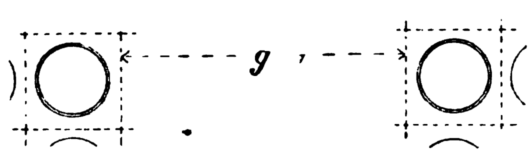


FIG. 435.

Let Σ (T) represent the sum of the end space required for the tubes. In fig. 434 this would be $2A+B$. Let D be the boiler diameter, then, $\Sigma (T) = 0{\cdot}45(D - d - K)^2$.

The value of K is found by multiplying the various water and steam spaces by the numbers contained in the following table and then adding them together :

Number of Furnaces in Boiler	Two	Three	Four
Multiply a by	·20	·70	·45
„ b „	·60	·60	·45
„ c „	...	·20	·70
„ the sum of e by	·30	·25	·35
„ „ „ g „	·20	·25	·35
„ f by	·40	·40	·50

The above formula will be found to agree fairly well with the general practice as long as h is equal to one-third of the boiler diameter ($\frac{1}{3}$ D). Where this relation does not exist, subtract from D, $\frac{3}{4}$ in., 1 in., and $1\frac{1}{8}$ in., for two-, three-, and four-furnaced boilers respectively, for every additional inch of steam space, so that if written out *in extenso* the above formula would be—

For two furnaces :

$$\Sigma(\mathrm{T}) = 0{\cdot}45\left(\frac{5}{4}\mathrm{D} - \frac{3}{4}h - d - {\cdot}20\left[a + 3b + 2f + \frac{3}{2}\,\Sigma(e) + \Sigma(g)\right]\right)^2.$$

For three furnaces :

$$\Sigma(\mathrm{T}) = 0{\cdot}45\left(\frac{4}{3}\mathrm{D} - h - d - {\cdot}7a + {\cdot}6b + {\cdot}2e + {\cdot}4f + {\cdot}25\left[\Sigma(e) + \Sigma(g)\right]\right)^2.$$

For four furnaces :

$$\Sigma(\mathrm{T}) = 0{\cdot}45\left(\frac{11}{8}\mathrm{D} - \frac{9}{8}h - d - {\cdot}45\,(a+b) + {\cdot}35\left[2c + \Sigma(e) + \Sigma(g)\right] + {\cdot}5f\right)^2.$$

The sign $\Sigma(\;)$ means that the letter in the bracket is the sum of the several dimensions.

These formulæ can be simplified by adopting the following average values : a and b vary from 3 ins. to 8 ins. and are usually 5 ins.

In two-furnaced boilers c varies from 3 ins. to 11 ins., but it is generally made $7\frac{1}{2}$ ins. In other boilers it varies from 8 to 12 ins., the mean being 10 ins.

e is sometimes as much as 15 ins. Generally both c and e are made 10 ins.

f and g both vary from 8 ins. to 12 ins. The usual practice is to make both about 10 ins.; this would give $10\frac{1}{2}$ ins. of clear water space at the wings, and 11 ins. at the centres.

The mean value of h is $\frac{1}{3}$ D.

Substituting these average values, the formulæ are reduced to the following :—

For two furnaces and two combustion chambers—

$$\Sigma(\mathrm{T}) = 0{\cdot}45\,(\mathrm{D} - d - 12\tfrac{1}{4})^2.$$

For three furnaces and three combustion chambers—

$$\Sigma(\mathrm{T}) = 0{\cdot}45\,(\mathrm{D} - d - 22\tfrac{1}{2})^2.$$

For four furnaces and two combustion chambers—

$$\Sigma(\mathrm{T}) = 0{\cdot}45\,(\mathrm{D} - d - 24\tfrac{3}{4})^2.$$

For four furnaces and three combustion chambers—

$$\Sigma(T) = 0{\cdot}45\,(D - d - 28\tfrac{1}{4})^2.$$

For the purpose of ascertaining the diameter it will be more convenient to alter these formulæ as follows :—

$$D = \tfrac{3}{2}\sqrt{\Sigma(T)} + d + K.$$

The Length of the Boiler is fixed by the length of the tubes or furnace, by the depth of the combustion chambers, and by the water space at their backs (see p. 92).

The latter should not be made less than 5 ins., but cases are met with where they are reduced to 3 ins. They should be made wider at the top than at the bottom, the usual angle being $\frac{1}{2}$ in. per foot of depth.

The Combustion Chambers, measured horizontally, should be made as deep as possible : 28 ins. and 36 ins. seem to be the smallest limits for single- or double-ended boilers respectively. Generally this depth is about 12 ins. greater than half the furnace diameter for single-ended boilers, while for double-ended ones with through combustion chambers it is made about 24 ins. deeper than half the furnace diameter.

The following are the relations usually existing between various boiler dimensions :—

Boilers with Two Furnaces.—The ratio of boiler diameter to furnace diameter is generally as 10 to 3, but sometimes 10 % more or less. Custom is equally divided between leading the two furnaces into two combustion chambers or into one. In the latter case the central water space between the tubes is sometimes dispensed with, but generally a few rows of tubes are left out along this line. The diameters of these boilers range from 8 ft. to 14 ft., and the lengths from 8 ft. to 10 ft. for natural draught, and up to 12 ft. for forced draught. The shortest double-ended boilers are 12 ft. long, and the longest 18 ft.

Boilers with Three Furnaces.—The ratio of boiler diameter to furnace diameter is generally as 4 to 1, but sometimes 7 % more or less. Generally three combustion chambers are fitted, but sometimes only one, and in that case the tubes are mostly divided into two groups with a water space in the centre. In rare cases there are no water spaces except at the sides. The diameters of these boilers range from 11 to 16 ft., and the lengths from 9 to 11 ft. for natural, and 12 ft. for forced draught. Double-ended boilers are sometimes made 20 ft. long.

Boilers with Four Furnaces.—The ratio of boiler diameter to furnace diameter is generally 5 to 1, but sometimes 5 % more or less. The combustion chambers are usually so arranged that the two central furnaces are led into one, and the two wing ones into separate chambers, so that there are one large and two small ones. Sometimes there are only two combustion chambers, and very rarely there are four, or only a single one. The diameters range from 13 to 17 ft., and the lengths are the same as for three-furnaced boilers. Double-ended ones are rarely built.

Boiler Performances.—The following are a few rough rules for

estimating the heating surface and power of a boiler under natural draught :—

H is the heating surface in square feet.
D is the boiler diameter in feet.
L is its length.
F is the sum of the furnace diameters in feet.
IHP is the expected horse-power.
WP is the working pressure.

$$H = D^2L.$$
$$H = \text{from } 100\,F \text{ to } 300\,F.$$
$$IHP = \text{from } \tfrac{1}{2}\,H \text{ to } \tfrac{1}{6}\,H.$$
$$\text{,,} = \text{,,} \ \tfrac{1}{2}\,D^2L \text{ to } \tfrac{1}{6}\,D^2L.$$
$$\text{,,} = \text{,,} \ 50\,F \text{ to } 100\,F.$$

Weight in tons $= D^2 \times L \times \cdot 0012$ for 180 lbs. WP.

Two lists of boiler performances have been published by F. Marshall, 'M. E.,' 1881, p. 449, and 1891, p. 337.

The following is a list of published drawings of marine boilers : -

F. Colyer, 1886. Three-furnaced oval boiler, by Maudslay, 12 ft. 4 ins. diam., 14 ft. 1 in. diam., 9 ft. 11 ins. long ; three-furnaced cylindrical boiler, 17 ft. 4 ins. diam., 9 ft. 1 in. long.

Schwarz Flemming, 1873. Forty-nine sketches of boilers.

C. Busley, 1883. Boilers in vessels of German Navy.

B. N. Bartol, 1851. Sketches and surfaces of American boilers.

J. T. Winton, 1883. Various types of boilers.

N. P. Burgh, 1873, contains drawings of about 20 boilers and sketches of 90 types of patented boilers from 1852–71.

N. Foley, 1891. Plates 6, 7, 8, give full detailed drawings of the three following boilers :

Single-ended, 2 furnaces, 39 ins. internal diam., 1,099 sq. ft. heating surface. Outside dimensions, 12 ft. × 9 ft. 8 ins. Weight, 25 tons. Steam pressure, 150 lbs.

Single-ended, 3 furnaces, $35\frac{1}{2}$ ins. internal diam., 1,334 sq. ft. heating surface. Outside dimensions, 12 ft. 6 ins. × 9 ft. 8 ins. Weight, 27 tons. Steam pressure, 150 lbs.

Double-ended, 4 furnaces, 34 ins. internal diam., 1,843 sq. ft. heating surface. Outside dimensions, 10 ft. 6 ins. × 16 ft. 6 ins. Weight, 34 tons. Steam pressure, 160 lbs. (See also p. 258.)

Other types of boilers will be found in the following lists :—

Navy Type. 'Surprise' and 'Alacrity,' Palmers', 'Enging,' vol. xl. pp. 447, 450 ; 'Melbourne,' Simmons & Co., 'Engr.' vol. lx. p. 392 ; 'Yorktown,' 'Enging.' vol. li. p. 493 ; 'Bergen,' 'Enging,' vol. xlix. p. 191 ; 'Mouche,' Belliss & Co., 'Enging.,' vol. xxxix. p. 81.

Locomotive Type.—Schichau, 'Enging.,' vol. xxxiv. p. 579 ; Yarrow, 'Enging.,' vol. xlii. p. 179 ; 'Sunderland,' Doxford, 'Enging.,' vol. xlix. p. 30 (for burning petroleum) ; Hick, Hargreaves, 'Enging.,' vol. xlix. p. 528 ; 'Barham' and 'Bellona,' Hawthorn, Leslie & Co., 'Enging.,' vol. l. p. 705 ; 'Phlegeton,' Soc. Ann. Claparede, 'Engr.,' vol. lx. p. 277 ; Sectional locomotive boiler for transport, Sandycroft Foundry, 'Enging.,' vol. xxxviii. p. 261 ; N. Foley, 1891, pl. x. and xi.,

Water Tube Boilers.—J. F. Spencer, 'M. E.,' 1859, p. 264 ; Z. Colburn, ibid., 1864, p. 61 ; Laybourne, ibid., 1871, p. 263 ; D. Joy, 'I. and S. I.,' 1874, p. 220 ; Perkins boiler, Maw, ciii. ; Adams and Co., 'Enging.,' vol. xxxiv. p. 251; J. F. Flanery, 'C. E.,' 1878, vol. liv. p. 123 ; J. T. Thornycroft, 'Enging.,' vol. xxxv. p. 463, vol. xliv. p. 105, vol. xlvii. p. 402 ; 'N. A.,' 1889, vol. xxx. p. 271 ; 'C. E.,' 1890, vol. xcix. p. 41 ; Ward's Patent, 'Enging.' vol. xlvii. p. 322 ; Yarrow, 'Enging.,' vol. li. p. 79. All recent vessels of the French Navy are having tubular boilers fitted (Belleville, Lagrafel-D'Allest), J. T. Milton, 'N. A.,' 1893, vol. xxxv.

In the following list the vessel's name and reference only are given. The drawings show the positions of boilers, pipes, &c., as fitted on board :—

W. H. Maw, 1883 : 'Gallia,' Plate xv. ; 'Hohenzollern,' xxi. ; H.M.S. 'Grappler,' 'Wrangler,' 'Wasp,' 'Banterer,' 'Espoir,' xlvii. and xlviii. ; 'Servia,' lxiv. ; H.M.S. 'Rover,' lxxi. ; 'Arizona,' xcii. ; 'Grecian,' cxviii. ; 'Assyrian Monarch,' cxlvi. and cxlvii. ; 'Czar,' cxxxi. ; H.M.S. 'Conqueror,' cxliii. ; 'Normandie,' clxviii.

'Engr.': 'Chicago,' vol. lxi. p. 86 ; 'Oroya,' vol. lxiii. p. 234 ; 'Grace Darling,' vol. lxv. p. 236 ; 'Elbe,' vol. lxiv. p. 522.

'Enging.' : 'Princess Elizabeth,' 'Princess Marie,' vol. xxxi. p. 119 ; 'Servia,' vol. xxxiii. p. 247 ; 'Satellite,' 'Conqueror,' vol. xxxv. pp. 266, 267 ; 'Czar,' vol. xxxv. p. 364 ; 'Churchill,' vol. xxxvii. p. 2 ; 'Normandie,' vol. xxxvii. p. 65 ; 'Godiva,' 'Stokesley,' and 'Hunstanton,' vol. xxxvii. p. 380 ; 'County of Salop,' vol. xxxviii. p. 516 ; H.M.S. 'Boadicea,' 'Bacchante,' vol. xl. pp. 325, 328 ; 'Alacrity' and 'Surprise,' vol. xl. p. 589 ; 'Kathleen Mavourneen,' vol. xli. p. 221 ; H.M.S. 'Mersey' and 'Rodney,' vol. xli. p. 449 ; 'Royal Prince,' vol. xli. p. 588 ; 'Westmoreland,' vol. xlii. p. 70 ; 'Gladiator,' vol. xliii. p. 104 ; 'County of York,' vol. xliii. p. 246 ; 'The Earl,' vol. xliv. p. 405 ; H.M.S. 'Orlando,' vol. xliv. p. 492 ; 'Elbe,' vol. xliv. pp. 656, 661 ; 'Islander,' vol. xlvi. p. 304 ; H.M.S. 'Barracouta,' vol. xlix. p. 476 ; 'Sunderland' (torpedo boat), vol. xlix. p. 32 ; 'Kaiser Wilhelm II.,' vol. l. p. 126 ; H.M.S. 'Barham' and 'Barracouta,' vol. l. p. 628 ; 'Normannia,' vol. l. p. 252 ; 'City of Vienna' (forced draught), vol. li. p. 398 ; 'Indra,' vol. li. p. 525 ; 'Scot,' vol. lii. p. 10 ; 'Ophir,' vol. lii. pp. 535, 591 ; H.M.S. 'Edgar,' vol. liii. p. 74.

The Scantlings of Boilers are in most cases determined either according to Lloyd's or the Board of Trade rules, to which a special chapter is devoted. Here it will not be out of place to indicate the methods to be employed for finding the best proportions, either as regards efficiency or weight. It is, of course, impossible to make any comparisons as regards cost, because the practices and available tools of different shops are not known ; nor will any attempt be made to deal with the subject exhaustively, and only two simple comparisons will be carried out. In the one case it will be shown which form of joint and which percentage of riveting gives the lightest boiler shell, but only if built strictly according to Lloyd's rules. In the other case the lightest means of staying flat boiler plates will be discussed, and a few remarks will be added, showing how to proceed when dealing with such rules.

List of Published Boiler Drawings.

Vessel's Name	Builder	Where Published[1]	Dimensions: Diameter Ft. Ins.	Dimensions: Length Ft. Ins.	Furnaces: No.	Furnaces: Type	Furnaces: Internal Diameter Inch	Working Pressure Lbs.	Grate Surface Sq. Ft.	Per Boiler: Heating Surface Sq. Ft.	Per Boiler: Indicated Horse-Power	—
DOUBLE-ENDED BOILERS.												
H.M.S. 'Edgar,' 'Hawk'	Fairfield Co.	'Enging.,' liii. p. 11	16 0	18 0	8	Fox	38	155	...	5027	...	...
'Hanoverian'	Doxford	Maw, cxix. / 'Enging.,' vol. xxxv. p. 338	15 6	20 0	6	Flanged	46	...	126	4015	...	...
'Ophir'	Napier	„ vol. lii. p. 586	15 1	19 1	6	Purves	42	160	126	4340	1400	Funnel 88 ft.
'Scot'	Denny	„ vol. lii. p. 39	15 3	18 2	6	Fox	45	170	139	3660	1943	Funnel 98 ft.
'Teucer,' 'Orestes,' &c.	Scott	Maw, clxii. / 'Enging.,' vol. xxxvi. p. 544	12 0 × 14 2	23 9	6	Fox	40	...	...	...	...	Holt type
'Parisian'	Napier	Maw, v. / 'Enging.,' vol. xxxii. p. 276	15 0	17 9	6	Plain	45	75	136	3794	1505	Funnel 72·5 ft.
'Mexican'	Clark	Maw, xxxiii. / 'Enging.,' vol. xxxiii. p. 51	12 8 × 16 6	17 6	6	Fox	40	90	...	3333	...	...
'Oroya'	Barrow	'Engr.,' vol. lxiii. p. 290	13 6	18 0	6	...	35	160	104	2940	...	...
'Kaiser W. II.'	Vulcan	'Enging.,' vol. l., p. 188	13 1½	19 1½	4	Fox	47	157	98	3000	1110	...
'Kathleen Mavourneen'	Jack	„ vol. xli. p. 271	14 3	16 1	6	Plain	40	...	110	3314	...	Funnel 62 ft.
'Arizona'	Elder	Maw, xcv.	13 6	18 0	6	Bowl rings	39	90	...	...	...	...
'Indra'	Fawcett Preston	'Enging.' vol. li. p. 525	13 6	16 6	6	Fox	38	150	...	3082	876	...
'Golconda' (*ex* 'Nulli Secunda')	Doxford	„ vol. xlii. p. 543	12 9	17 9	4	Fox	44	150	82	2844	...	Funnel 76
'Assyrian Monarch'	Earles	Maw, cli.	12 ·3	18 6	6	Fox	45	...	...	...	...	...
'Grecian'	Doxford	„ cxviii.	11 0 × 13 6	18 6	6	Plain	34	...	...	...	.	Funnel 61 ft.
...	German	'Engr.,' vol. lxiv. p. 516	12 6	17 5	4	Fox	43	143	...	...	...	...
'Tartar'	Richardson	„ vol. lxix. p. 420	13 0	14 9	4	..	42	...	80	2613	...	Funnel 69 ft. × 9½
H.M.S. 'Barracouta'	Palmers	„ vol. xlix. p. 476	10 9	17 6	4	Fox	37	...	...	...	...	...

SINGLE-ENDED BOILERS.

'Isle of Dursey'	Wallsend	'Enging.,' vol. xxxvii. p. 188	16 0	10 8	3	Plain	34	150	42	1650	600	...
'Ophir'	Napier	„ vol. lii. p. 587	15 1	10 8	3	Purves	42	...	63	2170	700	Funnel 88 ft.
'County of Salop'	Barrow	„ vol. xxxviii. p. 567	14 6	11 0	3	Fox	40	...	60	1872	557	...
'Main'	Palmers	'Engr.,' vol. lxix. p. 330	14 8	10 0	3	...	42	135	69	2350	1120	...
'Lynx,' 'Antelope,' &c.	Laird	'Enging.,' vol. xlix. p. 699	14 3	10 2	3	Purves	40	150	59	1900	850	...
I.R.N. 'Sinope'	Napier	„ vol. l. p. 81	14 7	9 10	3	Plain	48	125	76	1870	917	Funnel 80 ft.
...	Tod McGregor	Maw, viii.	12 6 × 15 8	10 3	3	Plain	39	...	...	...	...	...
'Hunstanton,' 'Godiva,' &c.	Westgarth English	'Enging.,' vol. xxxvii. p. 473	13 9	10 3	3	Plain	40	80	...	1798	700	..
'City of St. Francisco'	Roach	Maw, clvi.	13 0	10 6	3	Plain	39	..	...	...	...	
'Princess Elizabeth,' &c.	Elder	'Enging.,' vol. xxxi. p. 219	13 7½	9 1	3	Plain	54	70	91	1826	886	Furn'ce necks contracted
'Czar'	Wallsend	Maw, cxxxiii. 'Enging.,' vol. xxxv. p. 420	12 6	10 0	3	Flanged	39	80	58·5	1427	...	...
'Royal Prince'	Dickinson	„ vol. xli. p. 613	12 6	10 0	3	Fox	34	150	36	1350	409	Funnel 52 ft.
'The Earl'	Finch	„ vol. xliv. p. 403	12 6	10 0	3	...	36	100	54	...	530	Funnel 24 ft.
'County of York'	Barrow	„ vol. xliii. p. 297	12 0	10 4	3	Fox	33	164	50	1346	980	...
'Myra'	Dunlop	„ vol. liii. p. 351	...	...	2	Purves	42	...	...	...	...	...
'Dredger'	HawkesCrawshaw	„ vol. xxxv. p. 247	10 6 × 11 9	10 6	2	Plain	40	80	40	1150	300	...
'Westmoreland'	Rollo	„ vol. xlii. p. 71	12 0	9 0	3	Fox	32	150	34	1200	320	Funnel 44 ft.
'Normandy'	Elder	„ vol. xxxix. p. 283	11 10	8 3	2	Fox	46	110	49	1176	630	...
'Arabian'	Rankine & Blackmore	„ vol. xxxviii. p. 83	10 6	9 5	2	Fox	36	140	33	920	340	...
'Churchill'	Hall Russell	„ vol. xxxvii. p. 2	9 6	10 0	2	...	...	100	...	850	300	Funnel 32 ft.
'Grace Darling'	Fleming & Ferguson	'Engr.,' vol. lxv. p. 237	10 0	9 0	2	Purves	35	...	27	752	360	...
'Gladiator'	Ramage & Ferguson	'Enging.,' vol. xliii. p. 105	8 6	7 9	1	Plain	...	150	15	500	...	...
...	Hodge	„ vol. xxxvi. p. 4	4 9	6 9	1	Plain	24	...	...	...	...	...

[1] Maw, with Roman numerals, stands for W. H. Maw's *Recent Practice in Marine Engineering*, London, 1883; 'Engr.' stands for *Engineer*; and 'Enging.' for *Engineering*.

A very complete collection of rules and regulations by various Governments for boiler construction, &c., will be found in Delauny-Belville, 1886.

Riveted Joints.—Lap joints (fig. 436) and double butt-strap joints (fig. 437) are those most commonly used in boilers, though occasionally

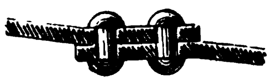


FIG. 436.

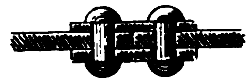


FIG. 437.

Rowe's joint (fig. 438) is met with ; it has been designed with the object of keeping the pitch of the outer row of rivets small, so as to increase the water-tightness. A similar object is evident in fig. 439.

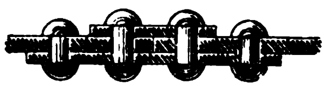


FIG. 438.

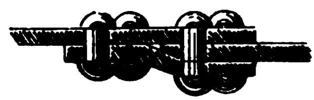


FIG. 439

Single butt-strapped joints (fig. 440) are only used on furnaces or to cover defective welds. Fig. 441 is a design intended to save weight.

When there are more than two rows of rivets in a lap joint, or more than three in a double butt-strapped one, it is desirable to arrange

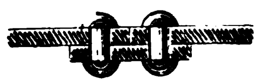


FIG. 440.

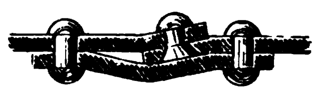


FIG. 441.

the rivets in such a manner that the percentage of strength of joint is the same for every row of rivets. It will be found that this is only the case in joints designed similarly to those shown in figs. 443 to 447. In these cases the quotient obtained by dividing the number of rivets in two adjoining rows is a constant value. If there are relatively fewer rivets in the inner row than in the outer one—as, for instance, in fig. 442—the percentage of strength along that line is too high ; while if there had been relatively more rivets, it would have been too low. Let the quotient of the number of rivets in one row, when divided by the number in the adjoining inner row, be denoted by m ; then B, the value of percentage of the joint—all rows having equal values—is found in the following table. A lower percentage than that given in this table should not be used, because the inner rows would be weaker than the outer ones, which gives a bad seam for caulking.

Table of Percentages of Riveted Joints.

m	$1\frac{1}{4}$	$1\frac{1}{3}$	$1\frac{1}{2}$	$1\frac{2}{3}$	$1\frac{3}{4}$	2	$2\frac{1}{4}$	$2\frac{1}{3}$
Lap joint, 3 rows .	...	52·6	63·6[1]	71·0	73·7	80·0	84·2	...
,, ,, 4 ,, .	53·0	60·8	71·4[2]	78·1	80 5	85·7	...	...
,, ,, 5 ,, .	60·2	68·2	78·4[3]	84·7	86·5	...	...	...
,, ,, 6 ,, .	65·6	73·8	82·6	87·9	...	...	...	...
,, ,, 7 ,, .	70·5	77·9	86·6	...	...	...	...	...
,, ,, 8 ,, .	74·2	81·2	...	...	...	...	...	...
Butt ,, 2 ,, .	...	...	55·5	63·9	67·3[4]	75·0[5]	80·2	87·1
,, ,, 3 ,, .	...	57·8	70·4	78·4	81·4[6]	87·5[7]	...	...
,, ,, 3 ,, .	$1\frac{1}{2}$ in each inner row		66·7	2 in each inner row		83·33[8]	...	

[1] See fig. 445. [2] See fig. 446. [3] See fig. 447.
[4] This is the value for Rowe's joint, having two rows of rivets on each side of butt with equal numbers of rivets in each (fig. 438). [5] See fig. 443.
[6] This is the value for Rowe's joint, having three rows of rivets on each side of butt, there being 1, 1, and $1\frac{2}{3}$ rivets respectively in the 1st, 2nd, and 3rd rows (fig. 448). [7] See fig. 444. [8] See fig. 442.

The riveted joint most commonly used is shown in fig. 442. It has been explained (see p. 165) that the distance from the edge of the plate to the circumference of the rivet hole should not be less than its

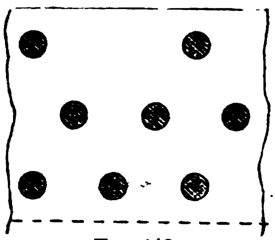


FIG. 442.

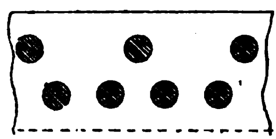


FIG. 443.

diameter. As regards the distances of the various rows of rivets from each other, a fairly uniform practice seems to exist of making the angular position of the rivets equal to about 50° to 55°, and in some works even 60° is customary. Some experiments (see p. 124) show that, except for an angle of about 60°, diagonal joints are no stronger than longitudinal ones. At this angle the longitudinal and diagonal pitches are equal. For smaller angles the percentage of the plate might be ascertained by measuring the zigzag line of all the outer rivets, subtracting their diameters, and then dividing by the zigzag length. Few joints would then show the high percentage now claimed for them.

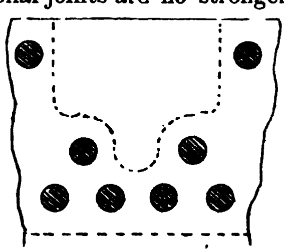


FIG. 444.

It has also been shown (see p. 168) that if the rows of rivets are placed very wide apart, the stresses are not uniformly distributed amongst them. This tends to lower the value of an ordinary joint, but the unequal straining would be somewhat reduced by cutting away the metal along the dotted line (fig. 444). In the absence of any conclusive experiments or inves-

tigations, it is safest to adhere to the general practice mentioned above. In double-ended boilers it will often be found that the percentage of plate remaining between the holes of the screwed stays is less than that between the rivets of the longitudinal seams. The stays will then have to be pitched diagonally. But some allowance might be made for the absence of bearing pressure (see p. 162).

Lightest Joints.—For the purpose of comparing the weight of various joints, and also the amount of metal removed by drilling, which is a sort of measure of the labour expended in the shops, the above drawings of riveted joints have been made; in every case the holes are placed at an angle of 60°. The dimensions are in accordance with Lloyd's Boiler Rule for a steel plate 1 in. thick, the rivet diameters not being less than 1 in., and the percentage of the plate and of the rivet sections being equal. The estimates are contained in the following table :—

Comparison of various Riveted Joints.

Type of Joint	Percentage	Rivets: No per Pitch	Rivets: Diam.	Rivets: Pitch	WP. of a 12-ft. Boiler	Weight per 1 ft. of Shell: 3 Joints including Rivet Heads	Weight per 1 ft. of Shell: Shell Plate and 3 Joints	Weight per 1 ft. of Shell: Weight of Rivets	Boiler Shell Weight / WP. × Length
			Inch	Inch	lbs.	lbs.	lbs.	lbs.	
Lap . fig. 445 .	70	3½	1	3·33	126	93	1643	39	13·0
„ . „ 446 .	77	5	1	4·35	142	114	1664	43	11·8
„ . „ 447 .	82·9	7½	1	5·84	150	174	1723	46	11·4
Butt . „ ([1]) .	74·5	2½	1	3·92	145	146	1696	37	11·7
„ . „ 443 .	77·8	3	1	4·50	151	155	1705	40	11·3
„ . „ 442 .	85·4	5	1	6·85	166	241	1791	42	10·8
„ . „ 444 .	89·1	7	1	9·16	173	306	1856	48	10·7
„ . „ 448 .	81·4	3¾	1·04	5·60	152	259	1809	39	11·9

[1] Like part of fig. 445.

The last column contains the quotients obtained by dividing the total weight of 12 ins. of length of the boiler shell by the respective working pressures, as found by the rules; these are convenient measures

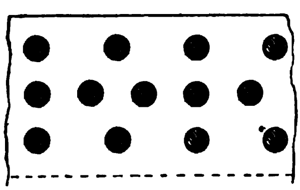


FIG. 445.

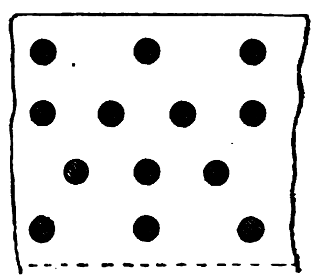


FIG. 446.

of the efficiencies of the various joints. It will be seen that the quadruple lap joint (fig. 446), though simpler, is almost as good as the

double-riveted butt joint (fig. 443), and that the common butt-strap joint (fig. 442) is nearly equal to the one shown in fig. 444. This order is slightly changed if the circumference of the shell is made up of only one or two instead of three plates. Of course, in works where the machines are incapable of dealing with plates of more than a certain

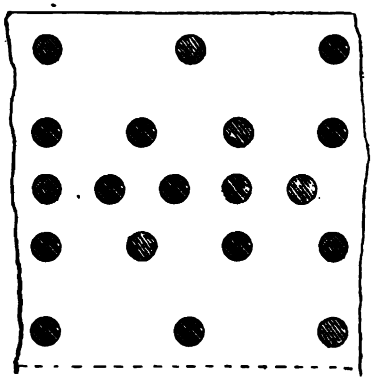


FIG. 447.

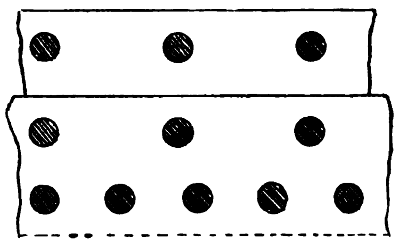


FIG. 448.

thickness it may be necessary to use joints of the very highest percentage, and then the one sketched in fig. 444 is undoubtedly the best (see p. 168).

Best Arrangement of Staying Flat Plates.—A comparison similar to the above can be carried out for flat plates. Here it cannot be a question of variation of pitch, for it is quite clear that the total weight of stays is a constant quantity, while the weight of the flat plates varies as the thickness, or inversely as the pitch of the stays, and by placing these very close together the plates could be made quite thin. In the following table the pitch of the stays in the combustion chamber plates is 10 ins. each way, the working pressure is 100 lbs., and the diameter of the screwed stay $1\frac{1}{2}$ in.

Table of Total Weight of Flat Combustion Chamber Plates.

—		Plate Thickness	Total Weight of 100 sq. ins., including Nuts, &c.
		Inch	lbs.
By Lloyd's Register Boiler Rules	stay ends riveted	·625	18·1
	„ „ nutted	·562	16·7
By Board of Trade Boiler Rules	„ „ riveted	·730	21·1
	„ „ nutted	·580	17·2

In the following table the area of steam-space plating supported by each stay of $2\frac{1}{4}$ ins. diameter is 300 square ins.

Table of Weight of Flat Steam-space Plates.

Conditions of Staying Flat Plates in Steam Space	Constants	Plate Thickness	Weight of 300 sq. ins. Plate	Weight of 300 sq. ins. Do. and Nuts, &c.
		Inch	lbs.	lbs.
Lloyd's Boiler Rules.				
Double nuts and no washers	175	·82	70·0	77·5
„ „ washers ($\frac{1}{3}$ P)	185	·80	68·0	78·5
„ „ „ riveted ($\frac{2}{5}$ P × $\frac{1}{2}$ T)	200	·77	65·4	84·7
„ „ „ „ ($\frac{2}{3}$ P × T)	220	·73	62·4	95·3
„ „ strip-riveted ($\frac{2}{3}$ P × T)	{220, 240}	·71	61·0	112·8
„ „ doubling plate ($\frac{2}{3}$ T)	175	{·61, ·41}	87·0	98·1
Board of Trade Rules.				
Stay ends nutted	112·5	·97	82·7	84·1
Double nuts and washers ($3d$ × $\frac{2}{3}$ T)	125	·91	78·0	91·7
„ „ „ riveted ($\frac{2}{3}$ P × T)	187·5	·73	62·6	95·6
„ „ strips „ „	200	·71	60·5	111·9

Here, again, it will be seen that the mode of attachment which permits of the use of the thinnest plate does not lead to the lightest construction.

Doubling strips and plates are sometimes arranged as shown in fig. 449, the adjoining plates overlapping each other. In such cases the corners should be cut away, so as to reduce the length of the taper. In the case illustrated in fig. 450, the plate has evidently been weakened by

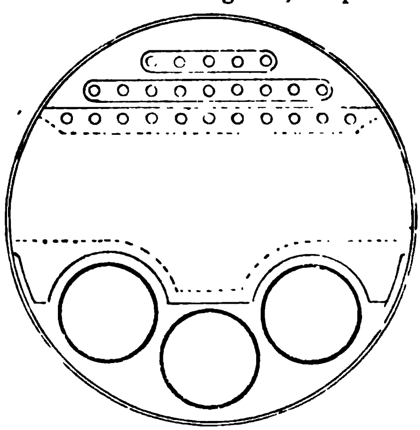


FIG. 449.

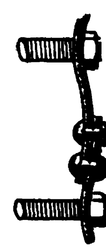


FIG. 450.

the seam. Doubling plates are also fitted to the front tube plate between the nests of tubes, and to the back plates of the boilers wherever the stays are too wide apart, and here, too, the laps are sometimes made very wide. Of course, the number or the sectional area of

the adjoining stays or stay tubes must be increased to bear the extra load. Angle irons are also fitted, but usually in such a manner that they give no support.

These few suggestions about the design of certain parts of boilers, so that they shall be as light as possible, can readily be extended to other parts, and also to the problem of cheapness and despatch; but when the main principles have been settled, care must still be exercised in carrying them out, for other circumstances may come into play. This occurs in the following simple design of a steel boiler in accordance with Lloyd's Register Boiler Rules. It is intended to illustrate how the various rules can be applied, and the various references to the tables will show how to use them advantageously.

In order to shorten the explanations, practically no notice has been taken of mechanical difficulties, which are discussed elsewhere, and the design is necessarily imperfect in these respects.

The External Diameter of the shell is not to exceed 13 ft. = 156 ins., the heating surface is to be 2,000 square ft., and the grate surface 50 square ft., but the length of the grate is to be limited to $5\frac{1}{4}$ ft. All the vertical water spaces are to be 10 ins. wide, and the steam-space stays are not to be placed closer together than 16 ins. The water space round each tube is to be 1 in., round the furnaces 4 ins., and above them $7\frac{1}{2}$ ins. Working pressure, 160 lbs.

The Furnace Diameter is found by dividing 50 square ft. by $5\frac{1}{4}$ ft., which gives 114 ins. of furnace fronts. Two furnaces would be too large, but three of 38 ins. internal and about $39\frac{1}{4}$ ins. external diameter will be convenient (see pp. 77, 255, and table, p. 286).

The Tube End Spaces have to be estimated, as explained on p. 253. The internal diameter of the shell is about 154 ins., and the water spaces, as explained on p. 253, are $9\frac{1}{2}$ ins. at the wings, 9 ins. between the nests of tubes, and 7 ins. above the furnaces. The steam space has been made one-third of the boiler diameter.

Water Spaces (p. 254)	*a*	*b*	*c*	*e*	*f*	*g*	Total
Water spaces, inches	4	4	7	7	9·5	9	—
Multiples	·7	·6	·2	·25	·4	2 × ·25	—
Products	2·8	2·4	1·4	1·7	3·8	4·5	16·6

Then Σ (T) = ·45 (154 − 39·25 − 16·6)² = 4,250 square ins. This is the available tube end area.

Tubes.—If the tubes are made $3\frac{1}{4}$ ins. diameter, then each one will occupy an end space of $(4\frac{1}{4})^2 = 18{\cdot}1$ square ins. Dividing this into 4,250 gives 235 as the necessary number of tubes. The tube surface will be about 80 per cent. of the total—say, 1,600 square ft.—and therefore each tube must have a surface of 6·8 square ft., which necessitates that its length should be 8 ft. (see table, p. 252). If 3-in. or $3\frac{1}{2}$-in. tubes had been decided upon, these numbers would have been respectively 268 and 7·6 ft. and 210 and 8·25 ft. These 235 tubes should be arranged in bundles of as nearly as possible equal numbers (78), and if possible in such a manner that the number of vertical and horizontal rows are all odd numbers, for then the staying is very much simplified.

The circular line A B (fig. 451) should be drawn; it marks off the boundary for the centres of the extreme tubes. Its radius is $66\frac{3}{8}$ ins. Measurement or calculation will show that after deducting the central water spaces there remain 121 ins. for the horizontal spacing of the tubes, which, as they are placed $4\frac{1}{4}$ ins. apart, number 28. The lines C D and E F, 7 ins. above the furnace crowns, indicate the lower boundary for the tube spaces, while N M is the upper limit. The height between these lines is $34\frac{1}{4}$ at the wings and 52 ins. at the centre, equal to about 8 and 12 tube pitches. The tube nests can now be arranged as shown in fig. 451, viz. $8 \times 10 + 12 \times 7 + 8 \times 10 = 244$ (odd number sbeing impossible). Of these, 6 fall away at the four corners, leaving 3 more than required. This agreement is sufficiently close for all practical purposes, but before proceeding it is advisable to estimate the

Heating Surface in the Combustion Chambers, so as to be sure that the total of 2,000 square ft. is reached. The clear depth, according to p. 255, is 31 ins., and the heating surface other than that of the tubes will be about 380 square ft., which, added to $238 \times 6{\cdot}8 = 1{,}620$, is exactly 2,000 square ft. Had this result not been obtained, then the tube lengths would have had to be altered a little.

The Lengths of the Shell Plates can now be fixed, and the riveted joints arranged.

The Thickness of the Front Tube Plate is determined either by the distances at M (fig. 451) across the wide water spaces, by the distance at N from the tubes to the shell, or by the mean pitch of the stay tubes. The latter dimension is the mean of the mean horizontal and of the vertical pitches, and is most easily determined by counting the number of tubes that form the four sides of a figure whose corners are occupied by stay tubes, then multiplying this sum by one-quarter of the tube pitches. In this design the tubes are placed $4\frac{1}{4}$ ins. apart, and the mean pitch can vary from $8\frac{1}{2}$ to $9\frac{9}{16}$, $10\frac{5}{8}$, $11\frac{11}{16}$, or $12\frac{3}{4}$ ins., requiring (see table, p. 292) that the plates should be either $\frac{5}{8}$, $\frac{11}{16}$, $\frac{3}{4}$, $\frac{13}{16}$, or $\frac{7}{8}$ in. thick. The plates across the $13\frac{1}{4}$ in. wide water spaces would have to be either $\frac{15}{16}$ or $\frac{7}{8}$ in. thick, according to the method of attaching the stay tubes. In this design the latter thickness will be used, and the stay tubes have therefore to be arranged as shown in fig. 451, those at the circumference being nutted.

If $\frac{5}{8}$-in. plates had been adopted it would have been necessary to fit about 25 extra stay tubes, as well as two doubling plates, to the front plate, as shown in dotted lines, whereby the cost would have been raised and no weight saved.

The Back Tube Plate need only be $\frac{13}{16}$ in. thick, as the mean pitch of the stay tubes does not exceed $11\frac{11}{16}$ ins.

Stay Tubes.—The constant for estimating the thickness of the front tube plate near the water line is 140, giving a pitch of 13·1 ins. with the $\frac{7}{8}$ in. plate. The top stay tubes will therefore have to support half this height and $1\frac{1}{2}$ pitch, equal to $6\frac{3}{8}$ in. below their centre line; but the area thus found has to be diminished by the section of four tubes: $(6{\cdot}55 + 6{\cdot}37) \times 8\frac{1}{2} - 4 \times 3\frac{1}{4}^{\circ} = 77$ square ins. The corner stays at M have each to support $(6{\cdot}55 + 4{\cdot}25) \times (4{\cdot}25 + 6{\cdot}75) - 2\frac{1}{4} \times 3\frac{1}{4}^{\circ} = 100$ square ins., and those in the nests of the tubes 87 square ins. The respective loads on each will be 12,300, 16,000, and 13,900 lbs., which,

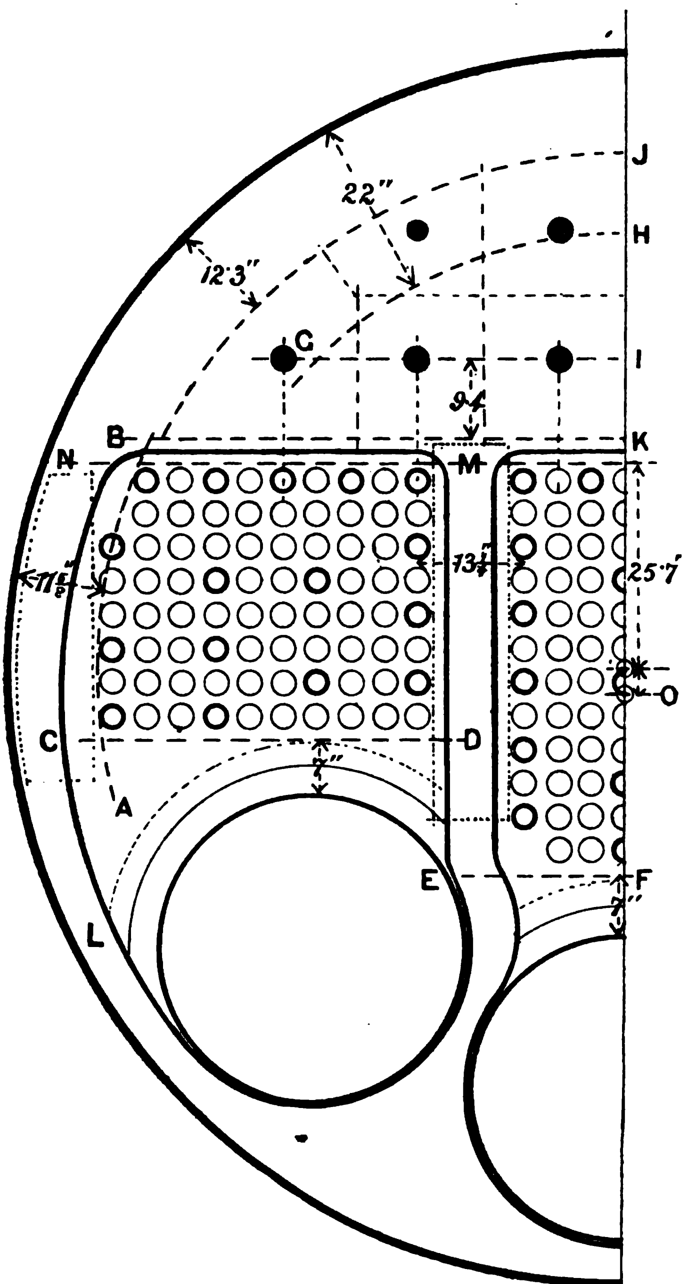


FIG. 451.

according to the table on p. 296, require that the minimum thicknesses of the stay tubes shall be $\frac{1}{4}$ in. and 9 threads per in., $\frac{5}{16}$ in. and 9 threads, and $\frac{1}{4}$ in. and 12 threads. The latter being too fine a screw, all the tubes will be made $\frac{5}{16}$ in. thick with 9 threads per inch.

End Plates in Steam Space.—One of the conditions of this design is, that the steam-space stays must be pitched at least 16 ins. apart. As previously mentioned (p. 262), the lightest construction is obtained if they are secured by double nuts. The constant being 175, 1-in. end plates will be required (see table, p. 290). But a trial shows that this leads to an inconvenient distribution of stays. Of the various alternatives which suggest themselves, a plate $1\frac{1}{8}$ in. thick will be used, provided that the flanging plant is sufficiently powerful to deal with it.

The maximum permissible pitch of stays is now 18·8 ins., and allowing $1\frac{1}{8}$ in. for the thickness of flange, and 2 ins. for its radius, the maximum distance from the shell at which the stays may be placed is 22 ins. This boundary line is marked G H (fig. 451). The lower boundary line, G I, is then drawn 15 ins. above the centres of the top row of tubes. This height is found by taking the mean of the pitches found for a $\frac{7}{8}$-in. plate with 140 constant and $1\frac{1}{8}$-in. plate with 175 constant. The position of the stays can now be marked off, care being taken that none of them are stationed over a water space, and all of them must fall outside of the line I G H.

The Sectional Areas of the Steam-space Stays are calculated by drawing the line B J, which is the lower limit of the area supported by the shell: it is found by adding half the permissible pitch of 18·8 ins. to the flange and its radius. B K is the upper limit of the area supported by the stay tubes, and, as already shown, it amounts to 6·55 ins. A few vertical and horizontal lines are now drawn to mark off the areas supported by the various stays, and it is then found that the load on the upper left-hand one is about 22,000 lbs., and on all the others about 53,000 lbs. According to the table on p. 295, these loads would require 2-in. and 3-in. stays with 6 threads per inch, because the next smaller sizes, $1\frac{3}{4}$ in. with 11 and $2\frac{7}{8}$ ins. with 12 threads, are not advisable.

The Combustion Chamber Side and Top Plates can now be drawn as shown in thick black lines. The centre of the wing radius may be placed at O, in order not to have an awkward corner at L, but then a stay will have to be fitted at A, which is inconvenient, especially if a manhole is fitted at the front end. The corners at M are struck with a radius of $3\frac{1}{2}$ ins., which allows 1 in. for the inner curve of the flange and $\frac{7}{8}$ in. for its thickness. For the position of seams see figs. 321–7, pp. 158 and 223.

Combustion Chamber Backs.—The centres of the furnaces, of the steam-space stays, and of the combustion chamber corners are transferred from fig. 451 to 452. The double boundary lines *a b c d* and *e f g l h* are then drawn. The outer one should be 3 ins. within the outer plates, so that the stay nuts, which are about 3 ins. over cants, will not have to rest on the curved part of the flange, which is assumed to be 1 inch. The inner boundary should be determined by theoretical considerations, (see p. 140), but as the numerical results are unpractical, the line has been drawn $1\frac{1}{2}$ ins. within the outer one, or $4\frac{1}{2}$ ins. from the side and top plates, so that the distance measured from the commencement of the flange to the edge of the nut is about $1\frac{1}{2}$ ins. All the stays at

the circumference should fall within these two lines, as shown in fig. 452.

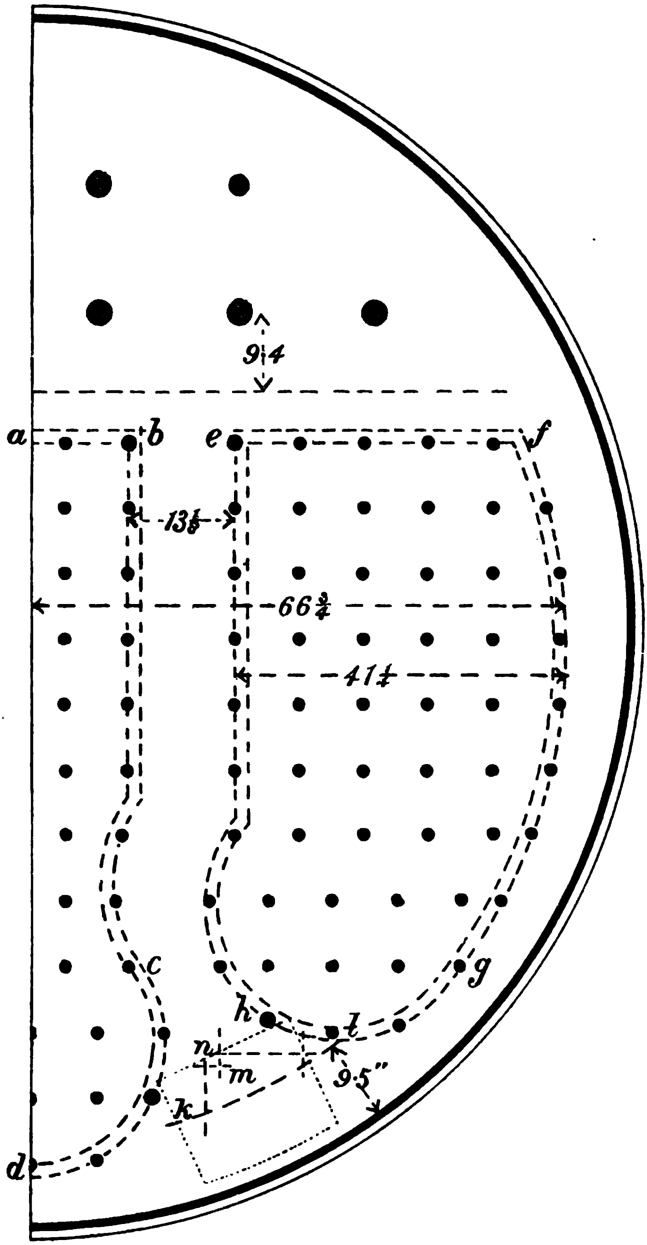


FIG. 452.

The maximum and minimum distances for the horizontal pitches of the screwed stays are $23\frac{1}{2}$ to $26\frac{1}{2}$ ins. for the centre and $38\frac{3}{4}$ to $41\frac{3}{4}$ ins. for the wing chambers. Using $\frac{9}{16}$ in. plates with nutted stays, the pitch is $8\frac{1}{4}$. They can be arranged as shown. It will be noticed that the lower rows have had to be shifted $4\frac{1}{8}$ in., and that the uppermost and lowermost stays just touch the inner boundary line.

Sectional Area of Screwed Stays.—The maximum permissible pitch for $\frac{9}{16}$-in. plates, viz. $8\frac{1}{4}$ ins., having been adopted, the table on p. 293 can be used with advantage, and the sizes of the stays are at once determined, viz. $1\frac{1}{2}$ in. diameter and 7 threads per inch. The size of the screwed stays of the circumferences are determined separately. The steam-space stays support the plating down to 9·4 ins. below the lowest ones, so that the top row of screwed stays has to support the remaining 6·1 ins. as well as $4\frac{1}{8}$ ins. lower down. Multiplying this sum by $8\frac{1}{4}$ ins. (the horizontal pitch) and 160 lbs., the load on each stay is found to be 13,500 lbs., which, according to the table on p. 295, requires a screw stay $1\frac{5}{8}$ in. diameter with 8 threads per inch.

The stays at *b* and *e* support $10\frac{1}{4}$ ins. $\times$ ($6\frac{3}{4}$ ins. + $4\frac{1}{8}$ ins.) $\times$ 160 = 17,800 lbs., and require stays of $1\frac{3}{4}$ in. diameter with 8 threads per inch. The others below these two, support loads of $10\frac{7}{8}$ ins. $\times$ $8\frac{1}{4}$ ins. $\times$ 160 = 14,360 lbs., and have to be $1\frac{5}{8}$ in. diameter, with at least 11 threads. This size will be adopted for all the stays at the circumference except *b* and *e*.

The Back End Plate.—It will be seen that the pitch of the stays across *b e* cannot be reduced below $13\frac{1}{8}$ ins. According to the table on p. 290, the thickness of the plate must be $\frac{15}{16}$ in., unless the stays at the edge of the combustion chamber are placed at an angle (see p. 231), or unless doubling strips are fitted between these stays. But neither alternative will be adopted.

The permissible distance from the upper screw stays to the lower steam-space stays has to be found. As drawn it measures $15\frac{1}{2}$ ins., and a calculation shows that, for the above thickness ($1\frac{1}{8}$ and $\frac{15}{16}$ in.) it might have been made $16\frac{1}{8}$ ins., showing that this part is strong enough (see p. 268).

Doubling Plates at Bottom of Back End.—By drawing the line *k l*, which is the width supported by the shell, and amounts to $9\frac{1}{2}$ ins., and also the boundaries *k m n* and *n l* of the areas supported by the lower screwed stays, it will be found that a small area is unsupported: a doubling plate has therefore to be fitted, as shown in dotted lines, and the stays near its corners at *k* and at *h* have to be increased to $1\frac{3}{4}$ in. diameter, in order to take their share of this extra load.

There are various other details—the shell (p. 261), the furnaces (p. 255), the girders, the combustion chamber side stays (pp. 29, 224), and the manholes (p. 274)—but they have either been discussed elsewhere or present no difficult features.

When they have been calculated, nothing remains to be done but to complete the drawing by inserting the various seams and flanges, and then making out a list of the full sizes of the plates (pp. 178, 206), the lengths of the stay bars, and the weights of the various sized rivets (p. 198), ready for ordering. All the dimensions should be carefully

checked, and if the boiler is to be built under the inspection of any society, the working pressure for each part should be calculated from the *invoiced dimensions* and with the help of the latest edition of their rules, and preferably without the use of any tables. If required, a tracing should also be submitted for approval, so as to obviate all risks of having to make alterations.

The two following tables, which contain the effective diameters and the effective sectional areas of screwed stays, may be of use when other stresses than those of the tables, pp. 294, 332, are to be adopted.

Effective Diameters of Screwed Stays.

Outside Diameters	Whitworth Screws		Number of Screw Threads per Inch						
			6	7	8	9	10	11	12
	Number of Threads		Effective Diameters at Bottom of Threads						
Inches		Inches	Inches	Inches	Inches	Inches	Inches	Inches	Inches
3/4	10	·62	...	...	...	...	·62	·63	·64
7/8	9	·73	...	...	...	·73	·75	·76	·76
1	8	·84	...	...	·84	·86	·87	·88	·89
1 1/8	7	·94	...	·94	·96	·98	1·00	1·01	1·02
1 1/4	7	1·07	...	1·07	1·09	1·11	1·12	1·13	1·14
1 3/8	6	1·16	1·16	1·19	1·21	1·23	1·25	1·26	1·27
1 1/2	6	1·29	1·29	1·32	1·34	1·36	1·37	1·38	1·39
1 5/8	5	1·37	1·41	1·44	1·46	1·48	1·50	1·51	1·52
1 3/4	5	1·49	1·54	1·57	1·59	1·61	1·62	1·63	1·64
1 7/8	4 1/2	1·59	1·66	1·69	1·71	1·73	1·75	1·76	1·77
2	4 1/2	1·72	1·79	1·82	1·84	1·86	1·87	1·88	1·89
2 1/8	4 1/2	1·84	1·91	1·94	1·96	1·98	2·00	2·01	2·02
2 1/4	4	1·93	2·04	2·07	2·09	2·11	2·12	2·13	2·14
2 3/8	4	2·05	2·16	2·19	2·21	2·23	2·25	2·26	2·27
2 1/2	4	2·18	2·29	2·32	2·34	2·36	2·37	2·38	2·39
2 5/8	4	2·30	2·41	2·44	2·46	2·48	2·50	2·51	2·52
2 3/4	4	2·43	2·54	2·57	2·59	2·61	2·62	2·63	2·64
2 7/8	3 1/2	2·51	2·66	2·69	2·71	2·73	2·75	2·76	2·77
3	3 1/2	2·63	2·79	2·82	2·84	2·86	2·87	2·88	2·89
3 1/8	...	...	2·91	2·94	2·96	2·98	3·00	3·01	3·02
3 1/4	3 1/4	2·86	3·04	3·07	3·09	3·11	3·12	3·13	3·14
3 3/8	...	...	3·16	3·19	3·21	3·23	3·25	3·26	3·27
3 1/2	3 1/4	3·11	3·29	3·32	3·34	3·36	3·37	3·38	3·39
3 5/8	...	...	3·41	3·44	3·46	3·48	3·50	3·51	3·52
3 3/4	3	3·32	3·54	3·57	3·59	3·61	3·62	3·63	3·64
3 7/8	...	...	3·66	3·69	3·71	3·73	3·75	3·76	3·77
4	3	3·57	3·79	3·82	3·84	3·86	3·87	3·88	3·89

Effective Sectional Areas of Stays.

Outside Diameters	Sectional Areas	Whitworth Screws		Number of Screw Threads per Inch						
				6	7	8	9	10	11	12
		Number of Threads		Effective Sectional Areas						
Inches	Sq. Ins.		Sq. Ins.	Sq. Ins.	Sq. Ins.	Sq. Ins.	Sq. Ins.	Sq. Ins.	Sq. Ins.	Sq. Ins.
3/4	·44	10	·30	...	...	...	...	·30	·31	·32
7/8	·60	9	·42	...	...	...	·42	·44	·45	·46
1	·79	8	·55	...	...	·55	·58	·60	·61	·63
1 1/8	·99	7	·70	...	·70	·73	·76	·78	·80	·81
1 1/4	1·23	7	·89	...	·89	·93	·96	·99	1·01	1·03
1 3/8	1·48	6	1·06	1·06	1·12	1·16	1·19	1·22	1·24	1·26
1 1/2	1·77	6	1·30	1·30	1·36	1·41	1·45	1·48	1·50	1·52
1 5/8	2·07	5	1·47	1·56	1·63	1·69	1·73	1·76	1·79	1·81
1 3/4	2·40	5	1·75	1·85	1·93	1·99	2·03	2·07	2·10	2·12
1 7/8	2·76	4 1/2	1·99	2·17	2·25	2·31	2·36	2·40	2·43	2·45
2	3·14	4 1/2	2·31	2·51	2·59	2·66	2·71	2·75	2·79	2·81
2 1/8	3·55	4 1/2	2·66	2·87	2·96	3·03	3·09	3·13	3·17	3·20
2 1/4	3·98	4	2·93	3·26	3·36	3·43	3·49	3·54	3·58	3·61
2 3/8	4·43	4	3·32	3·67	3·77	3·85	3·91	3·96	4·01	4·04
2 1/2	4·91	4	3·73	4·11	4·22	4·30	4·37	4·42	4·46	4·50
2 5/8	5·41	4	4·17	4·57	4·68	4·77	4·84	4·90	4·94	4·98
2 3/4	5·94	4	4·64	5·05	5·18	5·27	5·34	5·40	5·45	5·49
2 7/8	6·49	3 1/2	4·94	5·56	5·69	5·79	5·87	5·93	5·98	6·02
3	7·07	3 1/2	5·45	6·10	6·23	6·33	6·41	6·48	6·53	6·57
3 1/8	...	...	...	6·66	6·80	6·90	6·99	7·06	7·11	7·15
3 1/4	8·29	3 1/4	6·41	7·24	7·39	7·50	7·59	7·65	7·71	7·76
3 3/8	...	...	...	7·85	8·00	8·12	8·21	8·28	8·34	8·39
3 1/2	9·62	3 1/4	7·58	8·48	8·64	8·76	8·85	8·93	8·99	9·04
3 5/8	...	...	...	9·14	9·31	9·43	9·53	9·60	9·67	9·72
3 3/4	11·04	3	8·67	9·82	9·99	10·12	10·22	10·30	10·37	10·42
3 7/8	...	...	...	10·53	10·71	10·84	10·94	11·03	11·15	11·15
4	12·56	3	10·03	11·26	11·44	11·58	11·69	11·77	11·85	11·90

CHAPTER IX.

LLOYD'S REGISTER BOILER RULES.

EXTRACTS FROM RULES PUBLISHED IN JUNE 1893.

Rules for Determining the Working Pressure to be Allowed in New Boilers.

Cylindrical Shells of Iron Boilers.—The strength of circular shells of iron boilers to be calculated from the strength of the longitudinal joints by the following formula:—

$$\frac{C \times T \times B}{D} = \text{working pressure,}$$

where C = coefficient as per following table,
T = thickness of plate in inches,
D = mean diameter of shell in inches,
B = percentage of strength of joint found as follows—the least percentage to be taken.

For plate at joint $B = \frac{p-d}{p} \times 100$;

for rivets at joint $B = \frac{n \times a}{p \times T} \times 100$ with iron rivets in iron plates with punched holes,

$B = \frac{n \times a}{p \times T} \times 90$ with iron rivets in iron plates with drilled holes.

(In case of rivets being in double shear, $1{\cdot}75a$ is to be used instead of a),
where p = pitch of rivets,
d = diameter of rivets,
a = sectional area of rivets,
n = number of rows of rivets.

MEM.—In any case where the strength of the longitudinal joint is satisfactorily shown by experiment to be greater than given by this formula, the actual strength may be taken in the calculation.

Table of Coefficients.—Iron Boilers.

Description of Longitudinal Joint	For Plates $\frac{1}{2}$-in. thick and under	For Plates $\frac{3}{4}$-in. and above $\frac{1}{2}$-in.	For Plates above $\frac{3}{4}$-in. thick
Lap joint, punched holes . . .	155	165	170
" " drilled "	170	180	190
Double butt-strap joint, punched holes	170	180	190
" " " " drilled " .	180	190	200

NOTE.—The inside butt strap to be at least $\frac{3}{4}$ the thickness of the plate.

Cylindrical Shells of Steel Boilers.—The strength of cylindrical shells of steel boilers is to be calculated from the following formula :—

$$\frac{C \times (T-2) \times B}{D} = \text{working pressure in lbs. per square inch,}$$

where D = mean diameter of shell in inches,
T = thickness of plate in sixteenths of an inch,
C = 20 when the longitudinal seams are fitted with double butt straps of equal width,
C = 19·25 when they are fitted with double butt straps of unequal width, only covering on one side the reduced section of plate at the outer lines of rivets (fig. 438, p. 260),
C = 18·5 when the longitudinal seams are lap joints,
B = the least percentage of strength of longitudinal joint, found as follows :—

For plate at joint $B = \frac{p-d}{p} \times 100$;

for rivets at joint $B = \frac{n \times a}{p \times t} \times 85$ where steel rivets are used.

$B = \frac{n \times a}{p \times t} \times 70$ where iron rivets are used.

In case of rivets in double shear $1{\cdot}75a$ is to be used instead of a,
where p = pitch of rivets in inches,
t = thickness of plate in inches,
d = diameter of rivet holes in inches,
n = number of rivets used per pitch in the longitudinal joint,
a = sectional area of rivet in square inches.

Proper deductions are to be made for openings in shell.

All manholes in circular shells to be stiffened with compensating rings.

The shell plates under domes in boilers so fitted to be stayed from the top of the dome or otherwise stiffened.

NOTE.—The inside butt strap to be at least $\frac{3}{4}$ the thickness of the plate.

NOTE.—For the shell plates of superheaters or steam chests enclosed in the uptakes, or exposed to the direct action of the flame, the coefficients should be $\frac{2}{3}$ of those given above.

Stays.—The strength of stays supporting flat surfaces is to be cal-

culated from the smallest part of the stay or fastening, and the strain upon them is not to exceed the following limits, namely :—

Iron Stays.—For stays not exceeding $1\frac{1}{2}$ in. smallest diameter, and for all stays which are welded, 6,000 lbs. per sq. in.; for unwelded stays above $1\frac{1}{2}$ in. smallest diameter, 7,500 lbs. per sq. in.

Steel Stays.—For stays not exceeding $1\frac{1}{2}$ in. smallest diameter, 8,000 lbs. per sq. in.; for stays above $1\frac{1}{2}$ in. smallest diameter, 9,000 lbs. per sq. in. No steel stays are to be welded.

Stay Tubes.—The stress is not to exceed 7,500 lbs. per sq. in.

Flat Plates.—The strength of flat plates supported by stays to be taken from the following formula :—

$$\frac{C \times T^2}{P^2} = \text{working pressure in lbs. per square inch,}$$

where T = thickness of plate in sixteenths of an inch,

P = greatest pitch in inches,

C = 90 for iron or steel plates $\frac{7}{16}$ thick and under, fitted with screw stays with riveted heads,

C = 100 for iron or steel plates above $\frac{7}{16}$ thick, fitted with screw stays with riveted heads,

C = 110 for iron or steel plates $\frac{7}{16}$ thick and under, fitted with stays and nuts,

C = 120 for iron plates above $\frac{7}{16}$ thick, and for steel plates above $\frac{7}{16}$ and under $\frac{9}{16}$ thick, fitted with screw stays and nuts,

C = 135 for steel plates $\frac{9}{16}$ thick and above, fitted with screw stays and nuts,

C = 140 for iron plates fitted with stays with double nuts,

C = 150 for iron plates fitted with stays, with double nuts and washers outside the plates, of at least $\frac{1}{3}$ of the pitch in diameter and $\frac{1}{2}$ the thickness of the plates,

C = 160 for iron plates fitted with stays, with double nuts and washers riveted to the outside of the plates, of at least $\frac{2}{3}$ of the pitch in diameter and $\frac{1}{2}$ the thickness of the plates,

C = 175 for iron plates fitted with stays, with double nuts and washers riveted to the outside of the plates, when the washers are at least $\frac{2}{3}$ of the pitch in diameter and of the same thickness as the plates.

For iron plates fitted with stays, with double nuts and doubling strips riveted to the outside of the plates, of the same thickness as the plates, and of a width equal to $\frac{2}{3}$ the distance between the rows of stays, C may be taken as 175, if P is taken to be the distance between the rows, and 190 when P is taken to be the pitch between the stays in the rows.

For steel plates, other than those for combustion chambers, the values of C may be increased as follows :—

C = 140	increased to	175,
150	,,	185,
160	,,	200,
175	,,	220,
190	,,	240.

If flat plates are strengthened with doubling plates securely riveted to them, having a thickness of not less than $\frac{2}{3}$ of that of the plates, the strength to be taken from

$$\frac{C \times (T + \frac{t}{2})^2}{P^2} = \text{working pressure in lbs. per square inch ;}$$

where t = thickness of doubling plates in sixteenths, and C, T and P are as above.

NOTE.—In the case of front plates of boilers in the steam space these numbers should be reduced 20 %, unless the plates are guarded from the direct action of the heat.

For steel tube plates in the nest of tubes the strength to be taken from

$$\frac{140 \times T^2}{P^2} = \text{working pressure in lbs. per square inch ;}$$

where T = the thickness of the plates in sixteenths of an inch,
P = the *mean* pitch of stay tubes from centre to centre.

For the wide water spaces between the nests of tubes the strength to be taken from

$$\frac{C \times T^2}{P^2} = \text{working pressure in lbs. per square inch ;}$$

where P = the horizontal distance from centre to centre of the bounding rows of tubes, and
C = 120 where the stay tubes are pitched with two plain tubes between them, and are not fitted with nuts outside the plates,
C = 130 if they are fitted with nuts outside the plates,
C = 140 if each alternate tube is a stay tube not fitted with nuts,
C = 150 if they are fitted with nuts outside the plates,
C = 160 if every tube in these rows is a stay tube and not fitted with nuts,
C = 170 if every tube in these rows is a stay tube, and each alternate stay tube is fitted with nuts outside the plates.

The thickness of tube plates of combustion chambers in cases where the pressure on the top of the chambers is borne by these plates is not to be less than that given by the following rule :—

$$T = \frac{P \times W \times D}{1600 \times (D - d)} ;$$

where P = working pressure in lbs. per square inch,
W = width of combustion chamber over plates in inches,
D = horizontal pitch of tubes in inches,
d = inside diameter of plain tubes in inches,
T = thickness of tube plates in sixteenths of an inch.

Girders.—The strength of girders supporting the tops of combustion chambers and other flat surfaces to be taken from the following formula :—

$$\frac{C \times d^2 \times T}{(L-P) \times D \times L} = \text{working pressure in lbs. per square inch;}$$

where L = width between tube plates, or tube plate and back plate of chamber,
P = pitch of stays in girders,
D = distance from centre to centre of girder,
d = depth of girder at centre,
T = thickness of girder at centre. All these dimensions to be taken in inches.

Wrought Iron.

$$C = \begin{cases} 6{,}000 \text{ if there is one stay to each girder,} \\ 9{,}000 \text{ if there are two or three stays to each girder,} \\ 10{,}000 \text{ if there are four or five stays to each girder,} \\ 10{,}500 \text{ if there are six or seven stays to each girder,} \\ 10{,}800 \text{ if there are eight stays or above to each girder.} \end{cases}$$

Wrought Steel.

$$C = \begin{cases} 6{,}600 \text{ if there is one stay to each girder,} \\ 9{,}900 \text{ if there are two or three stays to each girder,} \\ 11{,}000 \text{ if there are four or five stays to each girder,} \\ 11{,}550 \text{ if there are six or seven stays to each girder,} \\ 11{,}880 \text{ if there are eight stays or above to each girder.} \end{cases}$$

Circular Furnaces.—The strength of plain furnaces to resist collapsing to be calculated from the following formula :—

$$\frac{89{,}600 \times T^2}{L \times D} = \text{working pressure in lbs. per square inch ;}$$

where T = thickness of plates in inches,
D = outside diameter of furnace in inches,
L = length of furnace in feet. If strengthening rings are fitted, the length between the rings is to be taken.

If the plates do not exceed $\frac{9}{16}$ in. in thickness, the pressure, however, is not to exceed

$$\frac{8{,}000 \times T}{D} = \text{lbs. per square inch.}$$

If the plates are of steel and exceed $\frac{9}{16}$ in. in thickness, the pressure is not to exceed

$$\frac{8{,}800 \times T}{D} = \text{lbs. per square inch.}$$

If the furnaces are fitted with a single Adamson's ring at about the middle of their length, the pressure may be calculated from

$$\frac{10{,}400 \times T}{D} = \text{working pressure in lbs. per square inch.}$$

If the furnaces are fitted with two Adamson's rings, then the pressure may be calculated from

$$\frac{11{,}400 \times T}{D} = \text{working pressure in lbs. per square inch.}$$

If the furnaces are fitted with a series of Adamson's rings at intervals not exceeding 23 ins., the pressure may be calculated from

$$\frac{1{,}000 \times (T-2)}{D} = \text{working pressure in lbs. per square inch;}$$

where T = thickness in sixteenths of an inch,
D = outside diameter of furnaces.

The strength of corrugated furnaces made of steel, having a less tensile strength than 26 tons per sq. in., the corrugations being 6 ins. apart and $1\frac{1}{2}$ in. deep, to be calculated from

$$\frac{1{,}000 \times (T-2)}{D} = \text{working pressure in lbs. per square inch.}$$

The strength of furnaces made of steel, having a tensile strength between 26 and 30 tons per sq. in., and corrugated on Fox's or Morison's plans, to be calculated from

$$\frac{1{,}259 \times (T-2)}{D} = \text{working pressure in lbs. per square inch.}$$

The strength of ribbed furnaces (with ribs 9 ins. apart) to be calculated from the following formula :—

$$\frac{1{,}160 \times (T-2)}{D} = \text{working pressure in lbs. per square inch.}$$

The strength of spirally corrugated furnaces to be calculated from the following formula :—

$$\frac{912 \times (T-2)}{D} = \text{working pressure in lbs. per square inch ;}$$

where T = thickness of plate in sixteenths of an inch,
and D = outside diameter of corrugated furnaces, or outside diameter of the ribbed furnaces, in inches.

The strength of Holmes' Patent Furnaces, in which the corrugations are not more than 16 ins. apart from centre to centre, and not less than 2 ins. high, to be calculated from the following formula :—

$$\text{Working pressure in lbs. per square inch} = \frac{945 \times (T-2)}{D} ;$$

where T = thickness of plain portions of furnace in sixteenths of an inch,
D = outside diameter of plain parts of the furnace in inches.

Donkey Boilers.—The iron used in the construction of the fire boxes, uptakes, and water tubes of donkey boilers shall be of good quality, and to the satisfaction of the surveyors, who may in any cases where they deem it advisable apply the following tests :—

Thickness of Plates	To Bend Cold through an Angle of	
	With the Grain	Across the Grain
$\frac{5}{16}$	80°	45°
$\frac{6}{16}$	70°	35°
$\frac{7}{16}$	55°	25°
$\frac{8}{16}$	40°	20°

The material to stand bending *hot* to an angle of 90°, over a radius not greater than 1½ time the thickness of the plates.

General Remarks about Boilers under Construction.—The surveyors will be guided in fixing the working pressure by the tables and formulæ annexed.

Any novelty in the construction of the machinery or boilers to be reported to the Committee.

The boilers, together with the machinery, to be inspected at different stages of construction.

The boilers to be tested by hydraulic pressure, in the presence of the engineer-surveyor, to twice the working pressure, and carefully gauged while under test.

Two safety valves to be fitted to each boiler and loaded to the working pressure in the presence of the surveyor. In the case of boilers of greater working pressure than 60 lbs. per sq. in., the safety valves may be loaded to 5 lbs. above the working pressure. If common valves are used, their combined areas to be at least half a square inch to each square foot of grate surface. If improved valves are used, they are to be tested under steam in the presence of the surveyor; the accumulation in no case to exceed 10% of the working pressure.

An approved safety valve also to be fitted to the superheater.

In winch boilers one safety valve will be allowed, provided its area be not less than half a square inch per square foot of grate surface.

Each valve to be arranged so that no extra load can be added when steam is up, and to be fitted with easing gear which must lift the valve itself. All safety-valve spindles to extend through the covers and to be fitted with sockets and cross handles, allowing them to be lifted and turned round in their seats, and their efficiency tested at any time.

Stop valves to be fitted so that each boiler can be worked separately.

Each boiler to be fitted with a separate steam-gauge, to accurately indicate the pressure.

Each boiler to be fitted with a blow-off cock independent of that on the vessel's outside plating.

The machinery and boilers are to be securely fixed to the vessel to the satisfaction of the surveyor.

Steel Boilers.—In cases where it is proposed to construct boilers of steel for classed vessels, or vessels intended for classification, the material is required to fulfil the following conditions:—

1. The material of stays and of plates is to have an ultimate tensile strength of not less than 26 and not more than 30 tons per sq. in. of section.

In all cases the ultimate elongation must not be less than 20% in a length of 8 ins.[1]

It is to be capable of being bent to a curve of which the inner radius is not greater than one and a half time the thickness of the

[1] Steel of a less tensile strength than 26 tons per sq. in., if satisfactory in other respects, may be allowed in any case where the scantlings are equal to those prescribed in the Rules for Iron Boilers. In such cases the surveyors should represent the facts for the Committee's consideration.

plates or bars, after having been heated uniformly to a low cherry-red, and quenched in water of 82° F.

2. Steel rivets are to be considered as part of the material, and, in addition to being subjected to a shearing test, they must be capable of withstanding the same tests as the plates are required to undergo.

3. Samples for testing are to be selected from each batch of plates submitted for approval, care being taken in the selection that, as far as possible, each cast or furnace charge from which the material has been produced is represented. In addition to these tests, the temper test is to be applied to samples taken from *every* plate intended to be used in the construction of boilers.

4. All the holes in steel boilers should be drilled, but if they be punched, the plates are to be afterwards annealed.

5. All plates that are dished or flanged, or in any way heated in the fire for working, except those that are subjected to a compressive stress only, are to be annealed after the operations are completed.

6. No steel stays are to be welded.

7. Unless otherwise specified, the rules for the construction of iron boilers will apply equally to boilers made of steel.

SUMMARY.

The preceding rules are summarised in the following short table, in which the method has been carried out of indicating all such dimensions as are measured in inches by capitals, such as are measured in sixteenths of an inch by small letters, and all coefficients, &c., by black letters :—

C and **C′** = coefficients.

W P = permissible working pressure.

B and **B′** = percentage of joint respectively of plate and of rivets.

N = number of rivets included within one pitch of external row.

T and t = thicknesses of plates measured respectively in inches and in sixteenths of an inch.

P and P′ = pitches of rivets of stays in flat plates in inches, or tubes in tube plate.

D = mean diameter of shells and diameter of furnaces in inches, measured as follows : for all plain furnaces, or made with ribs (Purves's), with flanges (Adamson's rings), or for Holmes's furnaces, the outside diameter of the plain cylindrical part is to be taken, and the thickness of the plates measured at these parts. For Fox's and Morison's corrugated furnaces the extreme external diameter is to be taken.

D and D′ = effective external and internal diameters of plain or stay tubes, and effective diameter of rivets, or of stays, in inches.

L′ = length of plain cylindrical parts of furnaces measured in feet.

L = length of girders measured in inches = internal distance between tube and back plates.

H = depth of girders measured in inches.

A = sectional area of stays or stay tubes, or of rivets, in square inches.

Σ **A** = sum of areas of holes in tube plate in square inches.

RIVETED JOINTS. (See appended Tables.)

Percentage of plate $\quad \mathbf{B} = \dfrac{P-D}{P}$.

Percentage of rivets $\quad \mathbf{B}' = \mathbf{C} \,.\, \mathbf{N} \,.\, \dfrac{A}{P \,.\, T}$, or $\mathbf{C}' \,.\, \mathbf{N} \,.\, \dfrac{D^2}{P \,.\, T}$.

TABLE OF COEFFICIENTS					
Materials of		Double Butt Straps		Lap Joints	
Plate	Rivet	C	C′	C	C′
Iron, punched .	Iron	175·0	137·4	100	78·5
„ drilled . .	„	157·5	123·8	90	70·7
Steel „ . . .	„	122·5	96·3	70	55·0
„ „ . .	Steel . . .	148·7	116·8	85	66·7

Iron BOILER SHELLS. $\quad \mathbf{W\,P} = \mathbf{C} \,.\, (\mathbf{B} \text{ or } \mathbf{B}') \,.\, \dfrac{T}{D}$, or $\mathbf{C}' \,.\, (\mathbf{B} \text{ or } \mathbf{B}') \,.\, \dfrac{t}{D}$.

TABLE OF COEFFICIENTS												
Joints	Double Butt Straps						Lap Joints					
Plates	C			C′			C			C′		
Thickness of Plates	½ in.	¾ in	above	in.	¾ in.	above	½ in.	¾ in.	above	½ in.	¾ in.	above
Punched .	170	180	190	10·61	11·24	11·87	155	165	170	9·68	10·31	10·61
Drilled .	180	190	200	11·24	11·87	12·50	170	180	190	10·61	11·24	11·87

Steel BOILER SHELLS. $\quad \mathbf{W\,P} = \mathbf{C} \,.\, (\mathbf{B} \text{ or } B') \,.\, \dfrac{t-2}{D}$.

TABLE OF COEFFICIENTS	C
Lap joints	18·5
Butt straps of unequal widths	19·25
„ of equal widths	20·0

FURNACES. (See appended Tables.)

Plain furnaces, $\quad \mathbf{W\,P} = 89{,}600 \,.\, \dfrac{T^2}{D \,.\, L}$, or $350 \,.\, \dfrac{t^2}{D \,.\, L}$.

The working pressure should not exceed $\mathbf{W\,P} = 8{,}000 \, \dfrac{T}{D}$, or $500 \, \dfrac{t}{D}$ for **iron** furnaces, or for **steel** ones up to $\frac{9}{16}$-inch plates;

$\mathbf{W\,P} = 8{,}800 \, \dfrac{T}{D}$, or $550 \, \dfrac{t}{D}$ for **steel** furnaces with plates above $\frac{9}{16}$ inch.

Furnaces with one Adamson's strengthening ring near middle,

$$\mathbf{W\,P} = 10{,}400 \,.\, \frac{T}{D}, \text{ or } 650 \,.\, \frac{t}{D}.$$

Furnaces with two Adamson's strengthening rings,

$$\mathbf{WP} = 11{,}400 \cdot \frac{T}{D}, \text{ or } 712{\cdot}5\,\frac{t}{D}.$$

PATENT AND OTHER FURNACES. $C' \cdot \frac{t-2}{D}.$

For measurements of D see p. 280.

TABLE OF COEFFICIENTS	C′
Corrugated flue, **steel** under 26 tons; or flues with Adamson's rings 23 inches apart	1,000
Corrugated flue, **steel** 26–30 tons	1,259
Purves's ribbed flue	1,160
Farnley's spirally corrugated flue	912
Holmes's flue	945

STAYED FLAT PLATES. $\mathbf{WP} = C \cdot \frac{t}{P}.$ (See appended Tables.)

TABLE OF COEFFICIENTS	Iron	Steel
Stay ends riveted plates up to $\frac{7}{16}$ inch	90	90
" " " above $\frac{7}{16}$ inch	100	100
" nutted " up to $\frac{7}{16}$ inch	110	110
" " " above $\frac{7}{16}$ inch	120	120
" " " $\frac{9}{16}$ inch thick and above	120	135
Double nuts	140	175
" and washers ($\frac{1}{3}$ P × $\frac{1}{2}$ T)	150	185
" and riveted washers ($\frac{2}{5}$ P × $\frac{1}{2}$ T)	160	200
" " ($\frac{2}{3}$ P × T)	175	220
" and doubling strips ($\frac{2}{3}$ P × T) lengthways	190	240
Tube plate	140	140
Tube plate between nests of tubes	Beaded	Nutted
When there are two plain tubes between stays	120	130
" is one plain tube between stays	140	150
" every tube is a stay tube	160	170
Doubling plates, $\mathbf{WP} = C\,\frac{(2t + t')}{2 \cdot P}$, t' is the thickness of the doubling plate in sixteenths of an inch		

$$\text{Tube plates, } \mathbf{WP} = \frac{1{,}600 \cdot (P - D') \cdot t}{L \cdot P}.$$

STAYS. (See appended Tables.)

$$\text{Stays.} \quad \mathbf{WP} = C \cdot \frac{A}{P \cdot P'}, \text{ or } C' \cdot \frac{D^2}{P \cdot P'}.$$

$$\text{Stay tubes.} \quad \mathbf{WP} = C \cdot \frac{A}{P \cdot P' - \Sigma A}, \text{ or } C' \cdot \frac{D^2 - D'^2}{P \cdot P' - \Sigma A}.$$

TABLE OF COEFFICIENTS	C		C′	
	Iron	Steel	Iron	Steel
Screwed stays up to $1\frac{1}{2}$ inch effective diameter .	6,000	8,000	4,712	6,283
„ above $1\frac{1}{2}$ inch effective diameter .	7,500	9,000	5,890	7,068
Stay tubes	7,500	7,500	5,890	5,890

GIRDERS. $$W P = C \frac{H^2 T}{L . (L-P) . P'}.$$

TABLE OF COEFFICIENTS	C	
Number of Stays per Girder	Iron	Steel
One	6,000	6,600
Two or three	9,000	9,900
Four or five	10,000	11,000
Six or seven	10,500	11,550
Eight or more	10,800	11,880

TABLES.

When used in the drawing office, it is strongly recommended that those parts of the following tables which are inapplicable to the particular works should be obliterated.

Table for Finding the Diameters of Rivets and of Pitches in Riveted Joints.

A short table will be found on p. 261 which shows the smallest permissible percentage for any form of joint, the condition being that when that particular percentage is adopted the strength of the joint shall be exactly the same for each row of rivets. If this percentage is exceeded, the inner rows will be stronger than the outer ones. If the joint is made of a smaller percentage, the inner rows of rivets will be the weakest and must be calculated separately.

Having decided on a particular percentage, the dimensions of the joint can be ascertained from the following table, which contains the values of $N . D \div T$ = number of rivets × diameter of rivets ÷ thickness of shell plate. Having found the number in the table, it is only necessary to multiply it by the thickness of the plate, and then, dividing by the number of rivets within one pitch, their diameters are found.

Example.—Butt-strap joint 1-inch steel plates, steel rivets, three rows, with two rivets in each inner row. Then the smallest percentage to be adopted for this joint is 83·33. Interpolating the values in the following table, it will be found that the value of $N . D \div T$ is 4·28. Dividing this by 5, the number of rivets, then their diameters must be ·856. In the first column of the same table will be found the value of pitch ÷ rivet diameter = 6. Therefore the pitch is 6 × ·856 = 5·136 inches.

If it is desired to make the rivet diameter equal to the thickness of the plate, then a value must be selected where $N . D \div T = 5$, i.e. $85\frac{5}{8}$ per cent.; then the pitch would have to be 6·98 inches.

In many of the high-percentage joints it will be found that unless the diameters of the rivets are less than the thickness of the plates the most efficient proportions cannot be adopted.

Pitch ÷ Rivet Diameter	Percentage of Plate	LAP JOINT: Punched Iron Plates, Iron Rivets	LAP JOINT: Drilled Iron Plates, Iron Rivets	LAP JOINT: Steel Plates, Steel Rivets	LAP JOINT: Steel Plates, Iron Rivets	BUTT STRAPS: Punched Iron Plates, Iron Rivets	BUTT STRAPS: Drilled Iron Plates, Iron Rivets	BUTT STRAPS: Steel Plates, Steel Rivets	BUTT STRAPS: Steel Plates, Iron Rivets
	Constants	100	90	85	70	$\frac{7}{4}$·100	$\frac{7}{4}$·90	$\frac{7}{4}$·85	$\frac{7}{4}$·70
2·50	·60	...	2·12	2·25	2·73	...	...	...	...
2·56	61	...	2·21	2·34	2·84	...	...	...	...
2·63	62	2·08	2·31	2·44	2·97	...	...	...	...
2·70	63	2·17	2·41	2·55	3·10	...	...	...	...
2·78	64	2·26	2·52	2·66	3·23	...	...	...	...
2·86	65	2·36	2·63	2·78	3·38	...	...	...	...
2·94	66	2·47	2·75	2·91	3·53	...	...	...	2·02
3·03	67	2·59	2·87	3·04	3·69	...	...	...	2·11
3·12	68	2·71	3·01	3·18	3·87	...	...	...	2·21
3·23	69	2·83	3·15	3·33	4·05	...	...	...	2·31
3·33	70	2·97	3·30	3·49	4·24	...	...	...	2·42
3·45	71	3·12	3·46	3·66	4·45	...	...	2·10	2·54
3·57	72	3·27	3·64	3·85	4·68	...	2·08	2·20	2·67
3·70	73	3·44	3·82	4·05	4·92	...	2·19	2·31	2·81
3·85	74	3·62	4·03	4·26	5·18	2·07	2·31	2·44	2·96
4·00	75	3·82	4·24	4·49	5·46	2·18	2·43	2·57	3·12
4·08	75½	3·92	4·36	4·61	5·61	2·24	2·49	2·64	3·20
4·17	76	4·03	4·48	4·74	5·76	2·30	2·56	2·71	3·29
4·26	76½	4·14	4·61	4·87	5·92	2·37	2·63	2·79	3·38
4·34	77	4·26	4·74	5·01	6·09	2·44	2·71	2·87	3·48
4·44	77½	4·38	4·87	5·16	6·27	2·51	2·78	2·95	3·58
4·55	78	4·51	5·02	5·31	6·45	2·58	2·86	3·03	3·68
4·65	78½	4·65	5·17	5·47	6·64	2·66	2·95	3·12	3·79
4·76	79	4·79	5·32	5·64	6·84	2·74	3·04	3·22	3·91
4·88	79½	4·94	5·49	5·81	7·05	2·82	3·13	3·32	4·03
5·00	80	5·09	5·66	5·99	7·28	2·91	3·23	3·42	4·16
5·13	80½	5·26	5·84	6·18	7·51	3·00	3·33	3·53	4·29
5·26	81	5·43	6·03	6·39	7·75	3·10	3·44	3·65	4·43
5·40	81½	5·61	6·23	6·60	8·01	3·21	3·56	3·77	4·58
5·56	82	5·80	6·44	6·82	8·29	3·32	3·68	3·90	4·73
5·71	82½	6·00	6·67	7·06	8·57	3·43	3·81	4·04	4·90
5·88	83	6·22	6·91	7·31	8·88	3·55	3·94	4·18	5·07
6·06	83½	6·44	7·16	7·58	9·20	3·68	4·08	4·33	5·26
6·25	84	6·68	7·43	7·86	9·54	3·82	4·25	4·49	5·46
6·45	84½	6·94	7·71	8·17	9·92	3·97	4·41	4·67	5·67
6·67	85	7·22	8·02	8·49	10·31	4·12	4·58	4·85	5·89
6·82	85⅓	7·41	8·23	8·72	...	4·23	4·70	4·98	6·05
6·98	85⅔	7·61	8·46	8·95	...	4·35	4·83	5·12	6·21
7·14	86	7·82	8·69	9·20	...	4·47	4·97	5·26	6·38
7·32	86⅓	8·04	8·94	9·46	...	4·60	5·11	5·41	6·57
7·50	86⅔	8·28	9·20	9·74	...	4·73	5·25	5·56	6·76
7·69	87	8·52	9·47	10·02	...	4·87	5·41	5·73	6·96
7·89	87⅓	8·78	9·75	...	...	5·02	5·57	5·90	7·17
8·10	87⅔	9·05	10·06	...	...	5·17	5·75	6·08	7·39
8·33	88	9·34	...	...	...	5·33	5·93	6·28	7·62
8·57	88⅓	9·64	...	...	...	5·51	6·12	6·48	7·87
8·82	88⅔	9·96	...	...	...	5·69	6·32	6·70	8·13
9·09	89	10·30	...	...	...	5·89	6·54	6·93	8·41
9·37	89⅓	...	...	...	...	6·09	6·77	7·17	8·70
9·68	89⅔	...	...	...	...	6·31	7·01	7·43	9·02
10·00	90	...	...	...	...	6·55	7·28	7·70	9·36

The following tables contain the internal diameters of furnaces, calculated according to Lloyd's Register Boiler Rules.

INTERNAL DIAMETERS OF CIRCULAR FURNACES IN INCHES.

Thickness, Inches	PLAIN OR FLANGED FLUES (See note at end of these Tables)									Steel Tenacity Above 26 T.		Under	Rings 23 ins. apart	Holmes
	$89{,}600 \cdot \frac{T^2}{D \cdot L}$, or $350 \cdot \frac{t^2}{D \cdot L}$					$C \cdot \frac{T}{D}$, or $C' \cdot \frac{t}{D}$				Corrug.	Purves	Corrug.		
										$C \times (t-2) \div D$				
L. or C.	9 ft.	8 ft.	7 ft.	6 ft.	5 ft.	8,000 500	8,800 550	10,400 650	11,400 712·5	1,259	1,160	1,000	1,000	945
60 lbs. Working Pressure.														
$\frac{3}{8}$	...	25·5	29·2	34·2	41·2	49·2	...	64·2	70·5	79·9	76·6	62·7	65·9	62·2
$\frac{13}{32}$	26·6	30·0	34·4	40·3	48·5	53·4	...	...	...	...	...	...	...	...
$\frac{7}{16}$	30·9	34·9	40·0	46·8	56·3	...	...	...	...	...	...	...	...	...
$\frac{15}{32}$	35·5	40·1	45·9	53·7	...	...	...	...	...	...	...	...	...	...
$\frac{1}{2}$	40·5	45·7	52·3	...	...	...	...	...	...	...	...	...	...	...
$\frac{17}{32}$	45·8	51·6	...	...	...	...	...	...	...	...	...	...	...	...
$\frac{9}{16}$	51·4	...	...	...	...	...	...	...	...	...	...	...	...	...
80 lbs. Working Pressure.														
$\frac{3}{8}$	...	...	...	25·5	30·7	36·7	...	48·0	52·7	58·9	57·2	46·0	49·2	46·5
$\frac{13}{32}$	...	...	25·6	30·0	36·2	39·8	...	51·7	...	...	...	52·2	55·4	52·3
$\frac{7}{16}$	...	25·9	29·7	34·8	42·0	42·9	...	...	...	...	...	...	...	...
$\frac{15}{32}$	26·3	29·8	34·2	40·1	45·9	45·9	...	...	...	...	...	...	...	...
$\frac{1}{2}$	30·1	34·0	39·0	45·7	49·0	49·0	...	...	...	...	...	...	...	...
$\frac{17}{32}$	34·1	38·4	44·1	51·6	52·1	52·1	...	...	...	...	...	...	...	...
$\frac{9}{16}$	38·3	43·2	49·5	...	...	...	...	...	...	...	...	...	...	...
$\frac{19}{32}$	42·7	48·2	55·2	...	...	...	..	...	...	...	...	...	...	...
$\frac{5}{8}$	47·3	53·4	...	...	...	...	...	...	...	...	...	...	...	...
$\frac{21}{32}$	52·3	...	...	...	...	...	...	...	...	...	...	...	...	...
100 lbs. Working Pressure.														
$\frac{3}{8}$	...	...	...	...	24·4	29·2	...	38·2	42·0	46·4	45·6	36·0	39·2	37·0
$\frac{13}{32}$	...	...	...	...	28·7	31·7	...	41·4	45·5	52·7	51·4	41·0	44·2	41·7
$\frac{7}{16}$	...	...	...	27·7	33·4	34·1	...	44·6	49·0	...	...	46·0	49·1	46·4
$\frac{15}{32}$	...	...	27·2	31·9	36·6	36·6	...	47·8	52·5	...	...	51·0	54·1	51·0
$\frac{1}{2}$	...	27·0	31·0	36·3	39·0	39·0	...	51·0	...	...	...	...	...	...
$\frac{17}{32}$	27·0	30·5	35·1	41·1	41·4	41·4	...	...	...	...	...	...	...	...
$\frac{9}{16}$	30·4	34·3	39·4	43·9	43·9	43·9	...	...	...	...	...	...	...	...
$\frac{19}{32}$	33·9	38·3	43·9	46·3 / 51·1	46·8 / 51·1	46·3	51·1	...	...	...	...	...	...	...
$\frac{5}{8}$	37·6	42·5	48·7	48·7 / 53·7	48·7 / 53·7	48·7	53·7	...	...	...	...	...	...	...
$\frac{21}{32}$	41·6	46·8	51·2 / 56·3	51·2 / 56·3	51·2 / 56·3	51·2	56·3	...	...	...	...	...	...	...
$\frac{11}{16}$	45·7	51·3	...	...	...	...	...	...	...	...	...	...	...	...
$\frac{23}{32}$	50·0	...	...	...	...	...	...	...	...	...	...	...	...	...
120 lbs. Working Pressure.														
$\frac{3}{8}$	...	...	...	...	...	24·2	...	31·7	34·9	38·0	37·9	29·3	32·6	30·7
$\frac{13}{32}$	...	...	...	...	...	26·3	...	34·4	37·5	43·2	42·7	33·5	36·7	34·6
$\frac{7}{16}$	...	...	...	...	27·7	28·3	...	37·0	40·7	48·5	47·5	37·7	40·8	38·5
$\frac{15}{32}$	...	...	...	26·4	30·3	30·3	...	39·7	43·6	53·7	52·2	41·8	44·9	42·4
$\frac{1}{2}$	...	...	25·7	30·1	32·3	32·3	...	42·3	46·5	...	...	46·0	49·0	46·2
$\frac{17}{32}$	...	25·3	29·0	34·1	34·4	34·4	...	45·0	49·4	...	...	50·2	53·1	50·1
$\frac{9}{16}$	25·1	28·4	32·6	36·4	36·4	36·4	...	47·6	52·3	...	...	...	...	...
$\frac{19}{32}$	28·1	31·7	36·4	38·4 / 42·4	38·4 / 42·4	38·4	42·4	50·3	...	...	...	...	...	...
$\frac{5}{8}$	31·2	35·2	40·4	40·4 / 44·7	40·4 / 44·7	40·4	44·7	...	...	...	...	...	...	...
$\frac{21}{32}$	34·4	38·9	42·4 / 46·8	42·4 / 46·8	42·4 / 46·8	42·4	46·8	...	...	...	...	...	...	...
$\frac{11}{16}$	37·8	42·7	44·5 / 49·0	44·5 / 49·0	44·5 / 49·0	44·5	49·0	...	...	...	...	...	...	...
$\frac{23}{32}$	41·4	46·5	46·5 / 51·3	46·5 / 51·3	46·5 / 51·3	46·5	51·3	...	...	...	...	...	...	...
$\frac{3}{4}$	45·2	48·5 / 53·6	48·5 / 53·6	48·5 / 53·6	48·5 / 53·6	48·5	53·6	...	...	...	...	...	...	...

INTERNAL DIAMETERS OF CIRCULAR FURNACES IN INCHES—*continued.*

Thickness. Inches	PLAIN OR FLANGED FLUES (See note at end of these Tables) $89{,}600 . \frac{T^2}{D . L}$, or $350 . \frac{t^2}{D . L}$					$C . \frac{T}{D}$, or $C' . \frac{t}{D}$				Steel Tenacity Above 26 T. Corrug.	Purves	Under Corrug.	Rings 23 ins. apart	Holmes
										$C \times (t - 2) \div D$				
L. or C.	9 ft.	8 ft.	7 ft.	6 ft.	5 ft.	8,000 / 500	8,800 / 550	10,400 / 650	11,400 / 712·5	1,259	1,160	1,000	1,000	945
							140 LBS. WORKING PRESSURE.							
3/8	...	...	...	...	...	...	...	27·1	29·8	32·0	32·4	24·6	27·8	26·2
13/32	...	...	...	...	...	...	...	29·4	32·3	36·5	36·5	28·1	31·3	29·6
7/16	...	...	...	...	...	24·1	...	31·6	34·7	41·0	40·6	31·7	34·8	32·9
15/32	...	...	...	...	25·6	25·9	...	33·9	37·1	45·5	44·6	35·3	38·3	36·2
1/2	...	...	...	25·7	27·6	27·6	...	36·1	39·7	50·0	48·7	38·9	41·9	39·5
17/32	...	...	24·7	29·0	29·3	29·3	...	38·4	42·2	54·4	52·8	42·4	45·4	42·8
9/16	...	24·2	27·8	31·0	31·0	31·0	...	40·7	44·7	...	...	46·0	48·9	46·1
19/32	...	27·0	31·0	32·7 / 36·1	32·7 / 36·1	32·7	36·1	42·9	47·2	...	...	49·6	52·4	49·4
5/8	26·6	30·0	34·4	34·5 / 38·0	34·5 / 38·0	34·5	38·0	45·2	49·6	...	...	53·1	...	53·7
21/32	29·3	33·1	36·2 / 38·1	36·2 / 39·9	36·2 / 39·9	36·2	39·9	47·4	52·1	...	...	...	...	...
11/16	32 2	36·4	37·9 / 41·8	37·9 / 41·8	37·9 / 41·8	37·9	41·8	49·6	...	...	...	...	...	...
23/32	35·3	39·6	39·6 / 43·7	39·6 / 43·7	39·6 / 43·7	39·6	43·7	51·9	...	...	...	...	...	...
3/4	38·5	41·4 / 43·5	41·4 / 45·6	41·4 / 45·6	41·4 / 45·6	41·4	45·6		...	...	...	...	...	...
							160 LBS. WORKING PRESSURE.							
3/8	...	...	...	...	...	...	...	...	26·0	27·5	28·2	...	24·2	...
13/32	...	...	...	...	...	...	...	25·6	28·1	31·4	31·8	24·1	27·3	25·8
7/16	...	...	...	...	...	...	...	27·6	30·3	35·3	35·4	27·2	30·4	28·7
15/32	...	...	...	...	...	...	...	29·5	32·5	39·3	38 9	30·4	33·4	31·5
1/2	...	...	...	...	24·0	24·0	...	31·5	34·6	43·2	42·5	33·5	36·5	34·4
17/32	...	...	...	25·3	25·5	25·5	...	33·5	36·8	47·1	46·1	36·6	39·5	37·3
9/16	...	...	24·2	27·0	27·0	27·0	...	35·4	39·0	51·1	49·6	39·7	42·6	40·2
19/32	...	...	27·0	28·5 / 31·5	28·5 / 31·5	28·5	31·5	37·4	41·1	55·0	53·2	42·9	45·7	43·1
5/8	...	26·1	30·0	30·0 / 33·1	30·0 / 33·1	30·0	33·1	39·4	43·3	...	...	46·0	48·7	46·0
21/32	25·5	28·8	31·5 / 33·1	31·5 / 34·8	31·5 / 34·8	31·5	34·8	41·3	45·4	...	...	49·1	51·8	48·9
11/16	28·0	31 7	33·0 / 36·4	33·0 / 36·4	33·0 / 36·4	33·0	36·4	43·3	47·6	...	...	52·2	...	51·8
23/32	30·7	34·5	34·5 / 38·0	34·5 / 38·0	34·5 / 38·0	34·5	38·0	45·3	49 8	...	...	...	...	...
3/4	33·5	36·0 / 37·9	36·0 / 39·7	36·0 / 39·7	36·0 / 39·7	36·0	39·7	47·2	52·0	...	...	...	...	...
							180 LBS. WORKING PRESSURE.							
3/8	...	...	...	...	...	...	...	...	...	24·0	25·0	...	...	...
13/32	...	...	...	...	...	...	...	...	24·9	27·5	28·2	...	24·2	...
7/16	...	...	...	...	...	...	...	24·4	26·8	31·0	31·3	...	26·9	25·4
15/32	...	...	...	...	...	...	...	26·2	28·7	34·5	34·5	26·6	29·6	27·9
1/2	...	...	...	...	...	...	...	27·9	30·7	38·0	37·7	29·3	32·3	30·5
17/32	...	...	...	...	...	...	...	29·6	32·6	41·5	40·8	32·1	35·0	33·1
9/16	...	...	...	24·9	24·9	24·9	...	31·4	34·5	45·0	44·0	34·9	37·8	35·7
19/32	...	...	24·9	25·2 / 27·8	25·2 / 27·8	25·2	27·8	33·1	36·4	48·5	47·1	37·7	40·5	38·2
5/8	...	...	26·5 / 28·4	26·5 / 29·3	26·5 / 29·3	26·5	29·3	34·9	38·3	51·9	50·3	40·4	43·2	40·8
21/32	...	25·5	27·7 / 29·8	27·7 / 30·8	27·7 / 30·8	27·7	30·8	36·6	40·2	...	...	43·2	45·9	43·4
11/16	24·8	28·2	29·2 / 32·2	29·2 / 32·2	29·2 / 32·2	29·2	32·2	38·3	42·2	...	...	46·0	48·6	45·9
23/32	27·1	30·5	30·5 / 33·7	30·5 / 33·7	30·5 / 33·7	30·5	33·7	40·1	44·1	...	...	48·8	51·3	48·5
3/4	29·7	31·8 / 33·5	31·8 / 35·2	31·8 / 35·2	31·8 / 35·2	31·8	35·2	41·8	46·0	...	...	51·6	...	51·0

Internal Diameters of Circular Furnaces in Inches—*continued.*

Thickness. Inches	PLAIN OR FLANGED FLUES (See note at end of these Tables) $89{,}600 . \frac{T^2}{D . L}$, or $350 . \frac{t^2}{D . L}$					$C . \frac{T}{D}$, or $C' . \frac{t}{D}$				**Steel** Tenacity Above 26 T. Corrug.	Above 26 T. Purves	Under Corrug.	Rings 23 ins. pitch	Holmes
										$C \times (t - 2) + D$				
L. or C.	9 ft.	8 ft.	7 ft.	6 ft.	5 ft.	8,000 / 500	8,800 / 550	10,400 / 650	11,400 / 712·5	1,259	1,160	1,000	1,000	945
200 lbs. Working Pressure.														
13/32	...	...	..	...	...	...	...	...	...	24·3	25·3	...	...	...
7/16	...	...	...	...	...	...	...	...	24·1	27·5	28·1	...	24·1	...
15/32	...	...	...	...	...	...	...	...	25·8	30·6	31·0	...	26·6	25·0
1/2	...	...	...	...	...	...	...	25·0	27·5	33·8	33·8	26·0	29·0	27·3
17/32	...	...	...	..	...	...	...	26·6	29·2	36·9	36·6	28·5	31·4	29 6
9/16	...	...	...	...	...	...	...	28·1	30·9	40·1	39·5	31 0	33·9	31·9
19/32	...	...	...	...	24·9	...	24·9	29·7	32·7	43·2	42·3	33·5	36·3	34·3
5/8	...	...	...	...	26 2	...	26·2	31·2	34·4	46 4	45·1	36·0	38·7	36·5
21/32	...	...	24·9 / 26·2	24·9 / 27·6	24·9 / 27·6	24·9	27·6	32·8	36·1	49·5	48·0	38·5	41·2	38·7
11/16	...	25·1	26·1 / 28·9	26·1 / 28·9	26·1 / 28·9	26·1	28·9	34·4	37·8	52·6	50·8	41·0	43·6	41·0
23/32	24·3	27·8 / 30·2	27·8 / 30·2	27·8 / 30·2	27·3 / 30·2	27·3	30·2	35·9	39·5	...	...	43·5	46·1	43·3
3/4	26·5	28·5 / 30·0	28·5 / 31·5	28·5 / 31·5	28·5 / 31·5	28·5	31·5	37·5	41·2	...	...	46·0	48·5	45·6

NOTE.—Of the small numbers in brackets, the upper ones are the internal diameters for iron furnaces, the lower ones for steel ones, estimated with the help of the formulæ (8,000 or 8,800) . T ÷ D.

Tables for Finding Maximum Pitch of Stays.

The following two sets of tables contain the maximum permissible pitches for various working pressures, thicknesses, and methods of attachments, both for stays and stay tubes. The mean pitch of stay tubes in an irregular square is found by adding together the four pitches, as measured horizontally and vertically, and dividing by four :—

Maximum Pitch in Inches of Stayed Flat Plates.

Plate Thickness	Stayed Iron Plates: Stay Ends Riveted	Stays Nutted	Stays with Double Nuts	Do. & Washers, ⅓ P × ½ T	Do. Washers, Riveted, ⅔ P × ⅔ T	Do. Do., ¾ P × T, or across Doubling Strips	Doubling Strips Lengthways	Stayed Steel Plates: Stay Ends Riveted	Stays Nutted	Stays with Double Nuts	Do. & Washers, ⅓ P × ½ T	Do. Riveted, ⅔ P × ⅔ T	Do. Do., ¾ P × T, or across Doubling Strips	Doubling Strips Longthways
Constants	90 100	110 120	140	150	160	175	190	90 100	110 120 135	175	185	200	220	240
60 lbs. Working Pressure.														
Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
$\frac{3}{8}$	7·34	8·12	9·16	9·48	9·79	10·2	10·6	7·34	8·12	10·2	10·5	10·9	11·4	12·0
$\frac{7}{16}$	8·57	9·47	10·7	11·0	11·4	11·9	12·4	8·57	9·47	11·9	12·2	12·7	13·4	14·0
$\frac{1}{2}$	10·2	11·3	12·2	12·6	13·0	13·6	14·2	10·2	11·3	13·6	14·0	14·6	15·3	16·0
$\frac{9}{16}$	11·5	12·7	13·7	14·2	14·6	15·3	16·0	11·5	13·5	15·3	15·8	16·4	17·2	18·0
$\frac{5}{8}$	12·8	14·1	15·2	15·8	16·3	17·0	17·7	12·8	15·0	17·0	17·5	18·2	19·1	20·0
$\frac{11}{16}$	14·1	15·5	16·8	17·3	17·9	18·7	19·5	14·1	16·5	18·7	19·3	20·1	21·1	...
$\frac{3}{4}$	15·4	16·9	18·3	18·9	19·5	20·4	21·2	15·4	18·0	20·4	21·1	...	...	...
$\frac{13}{16}$	16·7	18·3	19·8	20·6	21·2	...	...	16·7	19·5	...	...	...	...	...
$\frac{7}{8}$	18·0	19·7	21·3	...	...	...	...	18·0	21·0	...	...	...	...	...
$\frac{15}{16}$	19·3	21·1	...	...	...	...	...	19·3	...	...	...	...	...	...
1	20·6	...	...	...	...	...	...	20·6	...	...	...	...	...	...
80 lbs. Working Pressure.														
$\frac{3}{8}$	6·36	7·03	7·93	8·21	8·48	8·87	9·24	6·36	7·03	8·87	9·12	9·48	9·95	10·3
$\frac{7}{16}$	7·42	8·20	9·26	9·58	9·89	10·3	10·7	7·42	8·21	10·3	10·6	11·0	11·6	12·1
$\frac{1}{2}$	8·94	9·79	10·6	10·9	11·3	11·8	12·3	8·94	9·89	11·8	12·1	12·6	13·2	13·8
$\frac{9}{16}$	10·0	11·0	11·9	12·3	12·7	13·3	13·8	10·0	11·6	13·3	13·6	14·2	14·9	15·5
$\frac{5}{8}$	11·1	12·2	13·2	13·6	14·1	14·7	15·4	11·1	12·9	14·7	15·2	15·8	16·5	17·3
$\frac{11}{16}$	12·2	13·4	14·5	15·0	15·5	16·2	16·9	12·2	14·2	16·2	16·7	17·3	18·2	19·0
$\frac{3}{4}$	13·4	14·6	15·8	16·4	16·9	17·7	18·4	13·4	15·5	17·7	18·2	18·9	19·9	20·8
$\frac{13}{16}$	14·5	15·9	17·1	17·8	18·3	19·2	20·0	14·5	16·8	19·2	19·7	20·5	21·6	...
$\frac{7}{8}$	15·6	17·1	18·5	19·1	19·7	20·6	...	15·6	18·1	20·7	21·3	...	...	...
$\frac{15}{16}$	16·7	18·3	19·8	20·5	21·2	...	...	16·7	19·4	...	...	...	...	...
1	17·8	19·5	21·2	...	...	...	...	17·8	20·7	...	...	...	...	...
$1\frac{1}{16}$	19·0	20 8	...	...	...	...	...	19·0	...	...	...	...	...	...

Plate Thickness	STAYED **IRON** PLATES: Stay End Riveted	Stays Nutted	Stays with Double Nuts	Do. & Washers, ⅓P × ½T	Do. Washers, Riveted, ⅔P × ½T	Do. Do., ⅔P × T, or across Doubling Strips	Doubling Strips Lengthways	STAYED **STEEL** PLATES: Stay Ends Riveted	Stays Nutted	Stays with Double Nuts	Do. & Washers, ⅓P × ½T	Do. Riveted, ⅔P × ½T	Do. Do., ⅔P × T, or across Doubling Strips	Doubling Strips Lengthways
Constants	90 100	110 120	140	150	160	175	190	90 100	110 120 135	175	185	200	220	240
							100 lbs. Working Pressure.							
Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
3/8	5·79	6·29	7·09	7·34	7·58	7·93	8·27	5·79	6·29	7·93	8·16	8·48	8·89	9·29
7/16	6·64	7·34	8·28	8·57	8·85	9·26	9·64	6·64	7·34	9·26	9·52	9·89	10·3	10·8
1/2	8·00	8·76	9·46	9·79	10·1	10·6	11·0	8·00	8·76	10·6	10·8	11·3	11·8	12·3
9/16	9·00	9·85	10·6	11·0	11·3	11·9	12·4	9·00	10·4	11·9	12·2	12·7	13·3	13·9
5/8	10·0	10·9	11·8	12·2	12·6	13·2	13·7	10·0	11·6	13·2	13·6	14·1	14·8	15·4
11/16	11·0	12·0	13·0	13·4	13·9	14·5	15·1	11·0	12·7	14·5	14·9	15·5	16·3	17·0
3/4	12·0	13·1	14·1	14·6	15·1	15·8	16·5	12·0	13·9	15·8	16·3	16·9	17·7	18·5
13/16	13·0	14·2	15·3	15·9	16·4	17·1	17·9	13·0	15·1	17·1	17·6	18·3	19·2	20·1
7/8	14·0	15·3	16·5	17·1	17·7	18·5	19·2	14·0	16·2	18·5	19·0	19·7	20·7	...
15/16	15·0	16·4	17·7	18·3	18·9	19·8	20·6	15·0	17·4	19·8	20·4	21·1	...	...
1	16·0	17·5	18·9	19·5	20·2	21·1	...	16·0	18·5	21·1	...	...	...	...
1 1/16	17·0	18·6	20·1	20·8	...	...	...	17·0	19·7	...	...	...	...	...
1 1/8	18·0	19·7	...	...	...	...	...	18·0	20·9	...	...	...	...	...
							120 lbs. Working Pressure.							
3/8	5·19	5·74	6·48	6·70	6·92	7·24	7·55	5·19	5·74	7·24	7·44	7·74	8·12	8·48
7/16	6·06	6·70	7·56	7·82	8·07	8·45	8·80	6·06	6·70	8·45	8·69	9·03	9·47	9·89
1/2	7·30	8·00	8·64	8·94	9·23	9·66	10·0	7·30	8·00	9·66	9·93	10·3	10·8	11·3
9/16	8·21	9·00	9·72	10·0	10·3	10·8	11·3	8·21	9·54	10·8	11·1	11·6	12·1	12·7
5/8	9·12	10·0	10·8	11·1	11·5	12·0	12·5	9·12	10·6	12·0	12·4	12·9	13·5	14·1
11/16	10·0	11·0	11·8	12·2	12·7	13·2	13·8	10·0	11·6	13·2	13·6	14·2	14·8	15·5
3/4	10·9	12·0	12·9	13·4	13·8	14·4	15·1	10·9	12·7	14·4	14·8	15·4	16·2	16·9
13/16	11·8	13·0	14·0	14·5	15·0	15·7	16·3	11·8	13·7	15·7	16·1	16·7	17·6	18·3
7/8	12·7	14·0	15·1	15·6	16·1	16·9	17·6	12·7	14·8	16·9	17·4	18·0	18·9	19·7
15/16	13·6	15·0	16·2	16·7	17·3	18·1	18·8	13·6	15·9	18·1	18·6	19·3	20·3	21·1
1	14·6	16·0	17·2	17·8	18·4	19·3	20·0	14·6	16·9	19·3	19·8	20·6	...	...
1 1/16	15·5	17·0	18·3	19·0	19·6	20·5	...	15·5	18·0	20·5	21·0	...	...	...
1 1/8	16·4	18·0	19·4	20·1	20·8	...	...	16·4	19·0	...	...	...	...	...
							140 lbs. Working Pressure.							
3/8	4·81	5·31	6·00	6·21	6·41	6·70	7·02	4·81	5·31	6·70	6·89	7·17	7·51	7·85
7/16	5·61	6·20	7·00	7·24	7·48	7·82	8·19	5·61	6·20	7·82	8·04	8·36	8·76	9·16
1/2	6·76	7·40	8·00	8·28	8·55	8·94	9·36	6·76	7·40	8·94	9·19	9·56	10·0	10·5
9/16	7·60	8·32	9·00	9·31	9·62	10·0	10·5	7·60	8·83	10·0	10·3	10·7	11·2	11·8
5/8	8·45	9·25	10·0	10·3	10·6	11·1	11·7	8·45	9·81	11·1	11·4	11·9	12·5	13·1
11/16	9·29	10·2	11·0	11·3	11·7	12·2	12·8	9·29	10·8	12·2	12·6	13·1	13·7	14·4
3/4	10·1	11·1	12·0	12·4	12·8	13·4	14·0	10·1	11·7	13·4	13·8	14·3	15·0	15·7
13/16	10·9	12·0	13·0	13·4	13·8	14·5	15·2	10·9	12·7	14·5	14·9	15·5	16·2	17·0
7/8	11·8	12·9	14·0	14·4	14·9	15·6	16·3	11·8	13·7	15·6	16·0	16·7	17·5	18·3
15/16	12·6	13·8	15·0	15·5	16·0	16·7	17·5	12·6	14·7	16·7	17·2	17·9	18·7	19·6
1	13·5	14·8	16·0	16·5	17·1	17·8	18·7	13·5	15·7	17·8	18·3	19·1	20·0	20·9
1 1/16	14·3	15·7	17·0	17·5	18·1	19·0	19·9	14·3	16·6	19·0	19·5	20·3	...	...
1 1/8	15·2	16·6	18·0	18·6	19·2	20·1	21·1	15·2	17·6	20·1	20·6	...	...	...

Plate Thickness	STAYED IRON PLATES: Stay Ends Riveted	Stays Nutted	Stays with Double Nuts	Do. & Washers, $\frac{2}{3}$ P × $\frac{2}{3}$ T	Do. Washers, Rivetel, $\frac{2}{3}$ P × $\frac{2}{3}$ T	Do. Do., $\frac{2}{3}$ P × T, or across Doubling Strips	Doubling Strips Lengthways	STAYED STEEL PLATES: Stay Ends Riveted	Stays Nutted	Stays with Double Nuts	Do. & Washers, $\frac{2}{3}$ P × $\frac{2}{3}$ T	Do. Riveted, $\frac{2}{3}$ P × $\frac{2}{3}$ T	Do. Do., $\frac{2}{3}$ P × T, or across Doubling Strips	Doubling Strips Lengthways
Constants	90 100	110 120	140	150	160	175	190	90 100	110 120 135	175	185	200	220	240
160 lbs. Working Pressure.														
Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
$\frac{3}{8}$	4·50	4·97	5·61	5·80	6·00	6·27	6·53	4·50	4·97	6·27	6·45	6·70	7·03	7·34
$\frac{7}{16}$	5·25	5·80	6·54	6·77	7·00	7·31	7·62	5·25	5·80	7·31	7·52	7·82	8·20	8·57
$\frac{1}{2}$	6·32	6·93	7·48	7·74	8·00	8·36	8·71	6·32	6·93	8·36	8·60	8·94	9·38	9·80
$\frac{9}{16}$	7·11	7·79	8·41	8·71	9·00	9·41	9·80	7·11	8·26	9·41	9·67	10·0	10·5	11·0
$\frac{5}{8}$	7·90	8·66	9·35	9·68	10·0	10·4	10·8	7·90	9·18	10·4	10·7	11·1	11·7	12·2
$\frac{11}{16}$	8·69	9·53	10·2	10·6	11·0	11·5	11·9	8·69	10·1	11·5	11·8	12·2	12·9	13·4
$\frac{3}{4}$	9·48	10·3	11·2	11·6	12·0	12·5	13·0	9·48	11·0	12·5	12·9	13·4	14·0	14·6
$\frac{13}{16}$	10·2	11·2	12·1	12·5	13·0	13·5	14·1	10·2	11·9	13·5	13·9	14·5	15·2	15·9
$\frac{7}{8}$	11·0	12·1	13·0	13·5	14·0	14·6	15·2	11·0	12·8	14·6	15·0	15·6	16·4	17·1
$\frac{15}{16}$	11·8	12·9	14·0	14·5	15·0	15·6	16·3	11·8	13·7	15·6	16·1	16·7	17·5	18·3
1	12·6	13·8	14·9	15·4	16·0	16·7	17·4	12·6	14·7	16·7	17·2	17·8	18·7	19·5
$1\frac{1}{16}$	13·4	14·7	15·9	16·4	17·0	17·7	18·5	13·4	15·6	17·7	18·2	19·0	19·9	20·8
$1\frac{1}{8}$	14·2	15·5	16·8	17·4	18·0	18·8	19·6	14·2	16·5	18·8	19·3	20·1	21·0	...
180 lbs. Working Pressure.														
$\frac{3}{8}$	4·24	4·69	5·29	5·47	5·65	5·91	6·16	4·24	4·69	5·91	6·08	6·32	6·63	6·92
$\frac{7}{16}$	4·94	5·47	6·17	6·39	6·59	6·90	7·19	4·94	5·47	6·90	7·09	7·37	7·74	8·08
$\frac{1}{2}$	5·96	6·53	7·05	7·30	7·54	7·88	8·22	5·96	6·53	7·88	8·11	8·43	8·84	9·23
$\frac{9}{16}$	6·70	7·34	7·93	8·21	8·48	8·87	9·24	6·70	7·79	8·87	9·12	9·48	9·94	10·3
$\frac{5}{8}$	7·45	8·16	8·81	9·12	9·42	9·86	10·2	7·45	8·66	9·86	10·1	10·5	11·0	11·5
$\frac{11}{16}$	8·19	8·98	9·70	10·0	10·3	10·8	11·3	8·19	9·52	10·8	11·1	11·5	12·1	12·7
$\frac{3}{4}$	8·94	9·79	10·5	10·9	11·3	11·8	12·3	8·94	10·3	11·8	12·1	12·6	13·2	13·8
$\frac{13}{16}$	9·68	10·6	11·4	11·8	12·2	12·8	13·3	9·68	11·2	12·8	13·1	13·7	14·3	15·0
$\frac{7}{8}$	10·4	11·4	12·3	12·7	13·1	13·8	14·3	10·3	12·1	13·8	14·1	14·7	15·4	16·1
$\frac{15}{16}$	11·1	12·2	13·2	13·6	14·1	14·7	15·4	11·1	12·9	14·7	15·2	15·8	16·5	17·3
1	11·9	13·0	14·1	14·6	15·0	15·7	16·4	11·9	13·8	15·7	16·2	16·8	17·6	18·4
$1\frac{1}{16}$	12·6	13·8	14·9	15·5	16·0	16·7	17·4	12·6	14·7	16·7	17·2	17·9	18·7	19·6
$1\frac{1}{8}$	13·4	14·6	15·8	16·4	16·9	17·7	18·4	13·4	15·5	17·7	18·2	18·9	19·8	20·7
200 lbs. Working Pressure.														
$\frac{3}{8}$	4·02	4·44	5·02	5·19	5·36	5·61	5·84	4·02	4·44	5·61	5·77	6·00	6·29	6·57
$\frac{7}{16}$	4·69	5·19	5·85	6·06	6·26	6·24	6·82	4·69	5·19	6·54	6·73	7·00	7·34	7·66
$\frac{1}{2}$	5·65	6·19	6·69	6·92	7·15	7·48	7·79	5·65	6·19	7·48	7·69	8·00	8·39	8·76
$\frac{9}{16}$	6·36	6·97	7·53	7·79	8·05	8·41	8·77	6·36	7·39	8·41	8·65	9·00	9·43	9·85
$\frac{5}{8}$	7·07	7·74	8·36	8·66	8·94	9·35	9·74	7·07	8·21	9·35	9·61	10·0	10·4	10·9
$\frac{11}{16}$	7·77	8·52	9·20	9·52	9·83	10·2	10·7	7·77	9·03	10·2	10·5	11·0	11·5	12·0
$\frac{3}{4}$	8·48	9·29	10·0	10·3	10·7	11·2	11·6	8·48	9·85	11·2	11·5	12·0	12·5	13·1
$\frac{13}{16}$	9·19	10·0	10·8	11·2	11·6	12·1	12·6	9·19	10·6	12·1	12·5	13·0	13·6	14·2
$\frac{7}{8}$	9·89	10·8	11·7	12·1	12·5	13·0	13·6	9·89	11·5	13·0	13·4	14·0	14·6	15·3
$\frac{15}{16}$	10·6	11·6	12·5	12·9	13·4	14·0	14·6	10·6	12·3	14·0	14·4	15·0	15·7	16·4
1	11·3	12·3	13·3	13·8	14·3	14·9	15·5	11·3	13·1	14·9	15·3	16·0	16·7	17·5
$1\frac{1}{16}$	12·0	13·1	14·2	14·7	15·2	15·9	16·5	12·0	13·9	15·9	16·3	17·0	17·8	18·6
$1\frac{1}{8}$	12·7	13·9	15·0	15·5	16·1	16·8	17·5	12·7	14·7	16·8	17·3	18·0	19·8	19·7

Pitch in Inches of Tube Plate Stays, Centre to Centre.

Plate Thickness	Mean Pitch in Nest of Tubes	WIDE WATER SPACES						Mean Pitch in Nest of Tubes	WIDE WATER SPACES					
		Stay Ends Beaded			Stay Ends Nutted				Stay Ends Beaded			Stay Ends Nutted		
		Number of Plain Tubes between Stays							Number of Plain Tubes between Stays					
		0	1	2	0	1	2		0	1	2	0	1	2
Constants	140	160	140	120	170	150	130	140	160	140	120	170	150	130
	60 lbs. Working Pressure.							100 lbs. Working Pressure.						
$\frac{3}{8}$	9·16	9·80	9·16	8·48	10·1	9·49	8·83	7·10	7·59	7·10	6·57	7·82	7·35	6·84
$\frac{7}{16}$	10·7	11·4	10·7	9·90	11·8	11·1	10·3	8·28	8·85	8·28	7·67	9·13	8·57	7·98
$\frac{1}{2}$	12·2	13·1	12·2	11·3	13·5	12·6	11·8	9·47	10·1	9·47	8·76	10·4	9·80	9·12
$\frac{9}{16}$	13·7	14·7	13·7	12·7	15·1	14·2	13·2	10·6	11·4	10·6	9·86	11·7	11·0	10·3
$\frac{5}{8}$	15·3	16·3	15·3	14·1	16·8	15·8	14·7	11·8	12·6	11·8	11·0	13·0	12·2	11·4
$\frac{11}{16}$	16·8	18·0	16·8	15·6	18·5	17·4	16·2	13·0	13·9	13·0	12·0	14·3	13·5	12·5
$\frac{3}{4}$	18·3	19·6	18·3	17·0	20·2	19·0	17·7	14·2	15·2	14·2	13·1	15·6	14·7	13·7
$\frac{13}{16}$	19·9	21·2	19·9	18·4	...	20·6	19·1	15·4	16·4	15·4	14·2	16·9	15·9	14·8
$\frac{7}{8}$	21·4	...	21·4	19·8	...	...	20·6	16·6	17·7	16·6	15·3	18·3	17·1	16·0
$\frac{15}{16}$	...	...	...	21·1	...	...	...	17·7	19·0	17·7	16·4	19·6	18·4	17·1
1	...	...	...	...	...	...	...	18·9	20·2	18·9	17·5	20·9	19·6	18·2
$1\frac{1}{16}$	...	...	...	...	...	...	...	20·1	...	20·1	18·6	...	20·8	19·4
$1\frac{1}{8}$	...	...	...	...	...	...	...	...	...	...	19·7	...	...	20·5
	80 lbs. Working Pressure.							120 lbs. Working Pressure.						
$\frac{3}{8}$	7·94	8·49	7·94	7·35	8·75	8·22	7·65	6·48	6·93	6·48	6·00	7·14	6·71	6·24
$\frac{7}{16}$	9·26	9·90	9·26	8·57	10·2	9·58	8·92	7·56	8·07	7·56	7·00	8·33	7·83	7·29
$\frac{1}{2}$	10·7	11·3	10·7	9·79	11·7	11·0	10·2	8·64	9·24	8·64	8·00	9·52	8·94	8·33
$\frac{9}{16}$	11·9	12·7	11·9	11·0	13·1	12·3	11·5	9·72	10·4	9·72	9·00	10·7	10·1	9·37
$\frac{5}{8}$	13·2	14·1	13·2	12·2	14·6	13·7	12·7	10·8	11·5	10·8	10·0	11·9	11·2	10·4
$\frac{11}{16}$	14·6	15·6	14·6	13·5	16·0	15·1	14·0	11·9	12·7	11·9	11·0	13·1	12·3	11·4
$\frac{3}{4}$	15·9	17·0	15·9	14·7	17·5	16·4	15·3	13·0	13·9	13·0	12·0	14·3	13·4	12·5
$\frac{13}{16}$	17·2	18·4	17·2	15·9	18·9	17·8	16·6	14·0	15·0	14·0	13·0	15·5	14·5	13·5
$\frac{7}{8}$	18·5	19·8	18·5	17·1	20·4	19·2	17·8	15·1	16·2	15·1	14·0	16·7	15·7	14·6
$\frac{15}{16}$	19·8	21·2	19·8	18·4	...	20·5	19·1	16·2	17·3	16·2	15·0	17·9	16·8	15·6
1	21·2	..	2·12	19·6	...	...	20·4	17·3	18·5	17·3	16·0	19·0	17·9	16·7
$1\frac{1}{16}$	...	...	...	20·8	...	...	...	18·4	19·6	18·4	17·0	20·2	19·0	17·7
$1\frac{1}{8}$	...	...	...	...	...	...	...	19·4	20·8	19·4	18·0	...	20·1	18·7

Pitches in Inches of Tube Plate Stays, Centre to Centre.

Plate Thickness	Mean Pitch in Nest of Tubes	Wide Water Spaces						Mean Pitch in Nest of Tubes	Wide Water Spaces					
		Stay Ends Beaded			Stay Ends Nutted				Stay Ends Beaded			Stay Ends Nutted		
		Number of Plain Tubes between Stays							Number of Plain Tubes between Stays					
		0	1	2	0	1	2		0	1	2	0	1	2
Constants	140	160	140	120	170	150	130	140	160	140	120	170	150	130
	140 lbs. Working Pressure.							**180 lbs. Working Pressure.**						
$\frac{3}{8}$	6·00	6·41	6·00	5·55	6·61	6·21	5·82	5·29	5·66	5·29	4·90	5·83	5·48	5·10
$\frac{7}{16}$	7·00	7·48	7·00	6·48	7·71	7·25	6·74	6·17	6·60	6·17	5·72	6·80	6·39	5·95
$\frac{1}{2}$	8·00	8·56	8·00	7·41	8·81	8·28	7·71	7·05	7·54	7·05	6·53	7·77	7·30	6·80
$\frac{9}{16}$	9·00	9·62	9·00	8·32	9·91	9·32	8·67	7·94	8·48	7·94	7·35	8·75	8·22	7·65
$\frac{5}{8}$	10·0	10·7	10·0	9·26	11·0	10·4	9·64	8·82	9·43	8·82	8·16	9·72	9·13	8·50
$\frac{11}{16}$	11·0	11·8	11·0	10·2	12·1	11·4	10·6	9·70	10·4	9·70	8·98	10·7	10·0	9·35
$\frac{3}{4}$	12·0	12·8	12·0	11·1	13·2	12·4	11·6	10·6	11·3	10·6	9·78	11·7	11·0	10·2
$\frac{13}{16}$	13·0	13·9	13·0	12·0	14·3	13·5	12·5	11·5	12·3	11·5	10·6	12·6	11·9	11·0
$\frac{7}{8}$	14·0	15·0	14·0	13·0	15·4	14·5	13·5	12·3	13·2	12·3	11·4	13·6	12·8	11·9
$\frac{15}{16}$	15·0	16·0	15·0	13·9	16·5	15·5	14·5	13·2	14·1	13·2	12·2	14·6	13·7	12·7
1	16·0	17·1	16·0	14·8	17·6	16·6	15·4	14·1	15·1	14·1	13·1	15·5	14·6	13·6
$1\frac{1}{16}$	17·0	18·2	17·0	15·7	18·7	17·6	16·4	15·0	16·0	15·0	13·9	16·5	15·5	14·4
$1\frac{1}{8}$	18·0	19·3	18·0	16·7	19·8	18·6	17·3	15·9	17·0	15·9	14·7	17·5	16·4	15·3
	160 lbs. Working Pressure.							**200 lbs. Working Pressure.**						
$\frac{3}{8}$	5·61	6·00	5·61	5·20	6·18	5·81	5·41	5·02	5·37	5·02	4·65	5·53	5·20	4·84
$\frac{7}{16}$	6·55	7·00	6·55	6·07	7·22	6·78	6·31	5·86	6·26	5·86	5·42	6·45	6·06	5·64
$\frac{1}{2}$	7·48	8·00	7·48	6·93	8·25	7·74	7·21	6·69	7·15	6·69	6·20	7·39	6·93	6·45
$\frac{9}{16}$	8·42	9·00	8·42	7·80	9·28	8·71	8·11	7·53	8·05	7·93	6·97	8·31	7·80	7·26
$\frac{5}{8}$	9·35	10·0	9·35	8·67	10·3	9·68	9·01	8·37	8·94	8·37	7·75	9·22	8·66	8·06
$\frac{11}{16}$	10·3	11·0	10·3	9·53	11·3	10·6	9·92	9·20	9·84	9·20	8·52	10·1	9·52	8·87
$\frac{3}{4}$	11·2	12·0	11·2	10·4	12·4	11·6	10·8	10·0	10·7	10·0	9·29	11·1	10·4	9·67
$\frac{13}{16}$	12·2	13·0	12·2	11·3	13·4	12·6	11·7	10·9	11·6	10·9	10·1	12·0	11·3	10·5
$\frac{7}{8}$	13·1	14·0	13·1	12·1	14·4	13·6	12·6	11·7	12·5	11·7	10·8	12·9	12·1	11·3
$\frac{15}{16}$	14·0	15·0	14·0	13·0	15·5	14·4	13·5	12·5	13·4	12·5	11·6	13·8	13·0	12·1
1	15·0	16·0	15·0	13·9	16·5	15·5	14·4	13·4	14·3	13·4	12·4	14·7	13·9	12·9
$1\frac{1}{16}$	15·9	17·0	15·9	14·7	17·5	16·5	15·3	14·2	15·2	14·2	13·2	15·7	14·7	13·7
$1\frac{1}{8}$	16·8	18·0	16·8	15·6	18·6	17·4	16·2	15·1	16·1	15·1	13·9	16·6	15·6	14·5

Table of Relative Sizes of Screwed Stays and Flat Plates.

A simple relation exists between the thickness of a plate and its stays, and when the maximum pitch has been determined by the previous tables, the sizes, i.e. external diameters and pitch of threads, can be found direct by consulting the following table. Should the given sizes not be convenient, then either the diameter or the number of threads must be increased. Smaller numbers than those given in this table may not be adopted. The process can also be reversed.

Thickness of Plates in Inches		3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16	1	1 1/16	1 1/8
Method of Attaching Stays	Plate Con.	**IRON** PLATES AND **IRON** STAYS												
Stay ends riveted	90, 100	1 8	1⅛ 9	1⅜ 7	1½ 7	1⅝ 8	...	...	...	...	...	...	...	...
Stays nutted	110, 120	1⅛ 7	1¼ 7	1½ 6	1⅝ 7	1⅜ 11	...	...	...	...	...	...	...	...
Double nuts	140	1¼ 7	1⅜ 8	1½ 11	1⅝ 11	1¾ 6	1⅞ 7	2 8	2⅛ 10	2⅜ 6	2½ 7	2⅝ 8	2¾ 10	3 6
Do. and washers (⅓P × ½T)	150	1¼ 7	1⅜ 10	1⅝ 7	1⅝ 11	1¾ 9	1⅞ 10	2⅛ 6	2¼ 8	2⅜ 9	2½ 12	2¾ 7	2⅞ 8	3 10
Do. do. riveted (⅔P × ½T)	160	1¼ 9	1½ 6	1⅝ 9	1⅝ 11	1⅞ 6	2 7	2⅛ 9	2¼ 12	2½ 7	2⅝ 9	2¾ 11	3 7	3⅛ 8
Do. do. (⅔P × T)	175	1¼ 12	1½ 8	1⅝ 11	1¾ 6	1⅞ 7	2 10	2¼ 6	2⅜ 8	2½ 11	2¾ 7	2⅞ 9	3 12	3¼ 7
		STEEL PLATES AND **STEEL** STAYS												
Stay ends riveted	90, 100	⅞ 9	1 8	1⅛ 11	1¼ 12	1⅜ 12	1½ 12	1⅝ 11	...	...	...	...	...	...
Stays nutted	110, 120, 135	1 8	1⅛ 7	1¼ 9	1½ 7	1⅝ 8	1⅝ 12	1⅞ 6	...	...	...	...	...	...
Double nuts	175	1⅛ 11	1⅜ 7	1½ 8	1⅝ 11	1¾ 8	1⅞ 9	2 12	2¼ 7	2⅜ 8	2½ 9	2⅝ 12	2⅞ 7	3 8
Do. and washers (⅓P × ½T)	185	1¼ 7	1⅜ 8	1½ 10	1⅝ 11	1¾ 10	2 6	2⅛ 7	2¼ 9	2⅜ 12	2⅝ 7	2¾ 8	2⅞ 10	3⅛ 6
Do. do. riveted (⅔P × ½T)	200	1¼ 8	1⅜ 10	1⅝ 8	1⅝ 12	1⅞ 7	2 9	2⅛ 12	2⅜ 7	2½ 9	2¾ 6	2⅞ 7	3 10	3¼ 6
Do. do. (⅔P × T)	220	1¼ 10	1½ 7	1⅜ 10	1¾ 8	1⅞ 12	2⅛ 7	2¼ 10	2½ 7	2⅝ 9	2¾ 12	3 7	3⅛ 10	3⅜ 7
		STEEL PLATES AND **IRON** STAYS												
Stay ends riveted	90, 100	1 8	1⅛ 9	1⅜ 7	1½ 7	1⅝ 8	1⅝ 11	1¾ 7	...	...	...	...	...	...
Stays nutted	110, 120, 135	1⅛ 7	1¼ 7	1½ 6	1⅝ 11	1⅝ 12	1⅞ 7	2 7	...	...	...	...	...	...
Double nuts	175	1⅜ 6	1½ 9	1⅝ 11	1¾ 7	1⅞ 9	2 12	2¼ 7	2⅜ 10	2⅝ 6	2¾ 8	2⅞ 11	3⅛ 7	3¼ 9
Do. and washers (⅓P × ½T)	185	1⅜ 7	1½ 12	1⅝ 11	1¾ 9	2 6	2⅛ 8	2¼ 11	2½ 7	2⅝ 9	2⅞ 6	3 8	3⅛ 12	3⅜ 7
Do. do. riveted (⅔P × ½T)	200	1⅜ 10	1⅝ 7	1⅝ 11	1⅞ 6	2 8	2¼ 6	2⅜ 8	2⅝ 6	2¾ 8	2⅞ 12	3⅛ 8	3¼ 11	3½ 7
Do. do. (⅔P × T)	220	1½ 7	1⅝ 11	1¾ 7	1⅞ 10	2⅛ 7	2¼ 11	2½ 7	2⅝ 12	2⅞ 8	3⅛ 8	3¼ 9	3½ 6	3⅝ 9

Note.—The first of the numbers are the external diameters of the stays, the second numbers (black) are the number of threads per inch.

The following three tables contain respectively the maximum permissible load for screwed iron or steel stays, and for stay tubes, both for Whitworth screws and for finer pitches. They will be found convenient where the pitching of the stays is unequal :—

Permissible Working Loads on Iron Stays with Stresses of 6,000 and 7,500 lbs. per Square Inch.

Outside Diameters	Working Loads for Plus Threads	Whitworth Threads: Number of Threads	Whitworth Threads	Number of Threads per Inch: 6	7	8	9	10	11	12
			Working Loads of Screwed Stays							
Inches	Lbs.		Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
$\frac{3}{4}$	2,650	10	1,823	...	...	...	...	1,823	1,893	1,950
$\frac{7}{8}$	3,608	9	2,530	...	...	...	2,530	2,630	2,713	2,782
1	4,712	8	3,325	...	...	3,325	3,467	3,583	3,680	3,760
$1\frac{1}{8}$	5,964	7	4,183	...	4,183	4,388	4,552	4,684	4,795	4,887
$1\frac{1}{4}$	7,363	7	5,367	...	5,367	5,599	5,783	5,932	6,057	6,160
$1\frac{3}{8}$	8,909	6	6,359	6,359	6,697	6,956	7,162	7,328	7,466	7,580
$1\frac{1}{2}$	10,602	6	7,802	7,802	8,176	8,461	8,688	8,870	9,022	9,148
$1\frac{5}{8}$	15,550	5	8,831	9,391	9,802	10,114	10,361	10,561	13,407	13,579
$1\frac{3}{4}$	18,040	5	10,518	13,910	14,467	14,892	15,227	15,497	15,721	15,907
$1\frac{7}{8}$	20,710	$4\frac{1}{2}$	14,903	16,265	16,867	17,325	17,687	17,977	18,219	18,419
2	23,562	$4\frac{1}{2}$	17,338	18,804	19,449	19,943	20,331	20,643	20,901	21,115
$2\frac{1}{8}$	26,599	$4\frac{1}{2}$	19,952	21,527	22,219	22,744	23,159	23,491	23,767	23,995
$2\frac{1}{4}$	29,820	4	21,942	24,435	25,171	25,731	26,170	26,524	26,818	27,059
$2\frac{3}{8}$	33,225	4	24,876	27,526	28,308	28,900	29,366	29,740	30,052	30,307
$2\frac{1}{2}$	36,813	4	27,888	30,802	31,629	32,253	32,747	33,142	33,469	33,740
$2\frac{5}{8}$	40,591	4	31,295	34,260	35,133	35,792	36,310	36,727	37,072	37,357
$2\frac{3}{4}$	44,547	4	34,783	37,903	38,816	39,514	40,060	40,496	40,859	41,158
$2\frac{7}{8}$	48,689	$3\frac{1}{2}$	37,088	41,732	42,694	43,420	43,992	44,450	44,828	45,141
3	53,014	$3\frac{1}{2}$	40,874	45,744	46,751	47,511	48,107	48,586	48,984	49,309
$3\frac{1}{8}$	57,524	...	...	49,940	50,991	51,785	52,407	52,907	53,322	53,662
$3\frac{1}{4}$	62,229	$3\frac{1}{4}$	48,047	54,319	55,416	56,243	56,893	57,413	57,846	58,201
$3\frac{3}{8}$	67,097	...	...	58,883	60,025	60,885	61,561	62,103	62,551	62,920
$3\frac{1}{2}$	72,157	$3\frac{1}{4}$	56,827	63,631	64,817	65,711	66,413	66,978	67,442	67,825
$3\frac{5}{8}$	77,401	...	...	68,563	69,791	70,721	71,452	72,037	72,519	72,915
$3\frac{3}{4}$	82,830	3	65,059	73,680	74,946	75,916	76,670	77,277	77,777	78,189
$3\frac{7}{8}$	88,446	...	...	78,979	80,293	81,300	82,074	82,702	83,220	83,645
4	94,145	3	75,207	84 465	85,822	86,857	87,661	88,312	88,846	89,282

PERMISSIBLE WORKING LOADS ON **Steel** STAYS WITH STRESSES OF 8,000 AND 9,000 LBS. PER SQUARE INCH.

Outside Diameters	Working Loads for Plus Threads	Whitworth Threads: Number of Threads	Whitworth Threads: Working Loads of Screwed Stays	Number of Threads per Inch: 6	7	8	9	10	11	12
Inches	Lbs.		Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
3/4	3,534	10	2,431	...	...	...	...	2,431	·2,523	2,600
7/8	4,810	9	3,374	...	...	...	3,374	3,506	3,617	3,709
1	6,283	8	4,433	...	...	4,433	4,623	4,778	4,907	5,014
1 1/8	7,952	7	5,578	...	5,578	5,851	6,069	6,245	6,393	6,515
1 1/4	9,818	7	7,156	...	7,156	7,465	7,711	7,910	8,074	8,213
1 3/8	11,879	6	8,479	8,479	8,931	9,275	9,549	9,770	9,954	10,107
1 1/2	14,137	6	10,402	10,402	10,902	11,282	11,584	11,827	12,030	12,198
1 5/8	18,662	5	11,775	12,522	13,069	13,486	13,814	14,081	16,089	16,295
1 3/4	21,644	5	14,024	16,692	17,361	17,870	18,273	18,596	18,866	19,089
1 7/8	24,845	4 1/2	17,884	19,518	20,241	20,790	21,225	21,573	21,863	22,103
2	28,273	4 1/2	20,805	22,565	23,341	23,932	24,397	24,771	25,081	25,338
2 1/8	31,916	4 1/2	23,942	25,833	26,663	27,293	27,790	28,190	28,520	28,795
2 1/4	35,782	4	26,330	29,322	30,206	30,874	31,405	31,829	32,180	32,471
2 3/8	39,871	4	29,850	33,032	33,970	34,680	35,239	35,689	36,062	36,369
2 1/2	44,177	4	33,590	36,962	37,955	38,704	39,295	39,770	40,163	40,488
2 5/8	48,707	4	37,560	41,112	42,160	42,951	43,573	44,073	44,487	44,828
2 3/4	53,457	4	41,739	45,485	46,586	47,417	48,067	48,595	49,031	49,389
2 7/8	58,427	3 1/2	44,506	50,079	51,233	52,105	52,780	53,340	53,794	54,169
3	63,617	3 1/2	49,049	54,893	56,101	57,003	57,729	58,303	58,780	59,171
3 1/8	69,029	...	...	59,928	61,190	62,143	62,889	63,490	63,987	64,395
3 1/4	74,660	3 1/4	57,657	65,182	66,498	67,492	68,271	68,896	69,415	69,842
3 3/8	80,516	...	...	70,660	72,030	73,062	73,874	74,524	75,061	75,505
3 1/2	86,589	3 1/4	68,192	76,353	77,781	78,855	79,700	80,374	80,931	81,391
3 5/8	92,883	...	...	82,276	83,755	84,865	85,742	86,444	87,023	87,493
3 3/4	99,398	3	78,171	88,416	89,948	91,098	92,006	92,732	93,333	93,824
3 7/8	106,134	...	...	94,774	96,364	97,550	98,490	99,243	99,864	100,374
4	113,090	3	90,260	101,354	102 997	104,229	105,201	105,975	106,614	107,145

Stay Tubes, Sectional Areas, and Working Loads.

Number of Threads per Inch		8	9	10	11	12	Plus Thrds.	8	9	10	11	12	Plus Threads
External Diameter	Thickness of Metal	Sectional Area of Metal						Permissible Working Load (7,500 lbs.)					
Ins.	In.	Sq. In.	Sq. In.	Sq. In.	Sq. In.	Sq. In.	Sq. In.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
2	$\frac{1}{4}$	·89	·94	·98	1·02	1·05	1·37	6,690	7,080	7,380	7,650	7,870	10,310
	$\frac{5}{16}$	1·17	1·22	1·26	1·30	1·33	1·65	8,800	9,190	9,520	9,770	9,990	12,420
	$\frac{3}{8}$	1·43	1·48	1·52	1·56	1·59	1·91	10,740	11,130	11,430	11,790	11,920	14,360
$2\frac{1}{4}$	$\frac{1}{4}$	1·02	1·08	1·13	1·17	1·20	1·57	7,690	8,130	8,480	8,790	9,040	11,780
	$\frac{5}{16}$	1·35	1·41	1·46	1·50	1·53	1·94	10,180	10,620	10,970	11,270	11,520	14,260
	$\frac{3}{8}$	1·66	1·72	1·77	1·81	1·84	2·20	12,480	12,920	13,270	13,570	13,820	16,570
$2\frac{1}{2}$	$\frac{1}{4}$	1·16	1·22	1·27	1·32	1·35	1·76	8,680	9,180	9,570	9,910	10,190	13,230
	$\frac{5}{16}$	1·54	1·60	1·65	1·70	1·74	2·14	11,540	12,040	12,430	12,770	13,050	16,100
	$\frac{3}{8}$	1·89	1·96	2·01	2·05	2·09	2·50	14,210	14,710	15,100	15,440	15,720	18,770
$2\frac{3}{4}$	$\frac{1}{4}$	1·29	1·36	1·42	1·47	1·51	1·96	9,690	10,230	10,670	11,040	11,340	14,730
	$\frac{5}{16}$	1·72	1·79	1·85	1·90	1·94	2·39	12,920	13,470	13,890	14,270	14,580	17,940
	$\frac{3}{8}$	2·12	2·20	2·25	2·30	2·34	2·79	15,850	16,500	16,820	17,300	17,610	20,990
3	$\frac{1}{4}$	1·42	1·50	1·56	1·62	1·66	2·15	10,695	11,295	11,760	12,170	12,510	16,190
	$\frac{5}{16}$	1·90	1·98	2·04	2·10	2·14	2·63	14,280	14,880	15,360	15,760	16,080	19,780
	$\frac{3}{8}$	2·35	2·43	2·50	2·55	2·60	3·09	17,690	18,290	18,760	19,220	19,510	23,190
	$\frac{7}{16}$	2·78	2·86	2·93	2·98	3·03	3·52	20,910	21,510	21,990	22,390	22,710	26,410
$3\frac{1}{4}$	$\frac{1}{4}$	1·56	1·64	1·71	1·77	1·82	2·35	11,700	12,360	12,870	13,310	13,650	17,670
	$\frac{5}{16}$	2·08	2·17	2·24	2·30	2·35	2·88	15,650	16,310	16,820	17,260	17,610	21,630
	$\frac{3}{8}$	2·59	2·67	2·74	2·80	2·85	3·38	19,420	20,080	20,590	21,030	21,400	25,400
	$\frac{7}{16}$	3·06	3·15	3·22	3·28	3·33	3·86	23,020	23,670	24,180	24,630	24,990	29,000
$3\frac{1}{2}$	$\frac{1}{4}$	1·69	1·78	1·86	1·92	1·97	2·55	12,690	13,400	13,960	14,430	14,830	19,140
	$\frac{5}{16}$	2·27	2·36	2·43	2·50	2·55	3·12	17,020	17,730	18,290	18,760	19,160	23,460
	$\frac{3}{8}$	2·82	2·91	2·99	3·05	3·10	3·68	21,170	21,870	22,440	22,910	23,280	27,390
	$\frac{7}{16}$	3·35	3·44	3·51	3·58	3·63	4·20	25,120	25,830	26,390	26,860	27,230	31,560
$3\frac{3}{4}$	$\frac{1}{4}$	1·82	1·92	2·00	2·07	2·13	2·74	13,690	14,440	15,060	15,570	15,960	20,610
	$\frac{5}{16}$	2·45	2·55	2·63	2·70	2·75	3·37	18,390	19,150	19,750	20,260	20,660	25,310
	$\frac{3}{8}$	3·05	3·15	3·23	3·30	3·36	3·97	22,890	23,660	24,260	24,770	25,200	29,820
	$\frac{7}{16}$	3·63	3·73	3·81	3·88	3·93	4·55	27,220	27,990	28,590	29,100	29,520	34,140

CHAPTER X.

BOARD OF TRADE BOILER RULES.

EXTRACTS FROM THE 'INSTRUCTIONS AS TO THE SURVEY OF THE MACHINERY OF STEAM SHIPS.' PUBLISHED BY THE BOARD OF TRADE, FEBRUARY 1893.

Iron Boilers and Superheaters.

The Surveyor to fix Pressures on Safety Valves.—80.[1] The surveyor is required by the Act to fix the limits of weight to be placed on the safety valves of passenger steam ships. In performing this very responsible and onerous duty he must be very careful, as in the event of accident it will be necessary for him to satisfy the Board of Trade that he used due caution. On the one hand he must be careful as regards safety, and on the other hand he must not unduly reduce the pressure on a boiler. The surveyor himself having fixed the limits of the weight, is then required to declare that in his judgment the boiler and machinery are sufficient for the service intended and in good condition, and that they will be sufficient for twelve months or such other period as he may, in his judgment, determine. For his guidance the following suggestions are given, and he should not depart from them in any case without first reporting particulars to the Board of Trade, and asking for instructions.

Working Pressure to be fixed by Calculation.—81. The surveyor should fix the working pressure for boilers by a series of calculations of the strength of the various parts, and according to the workmanship and material. The Board of Trade, upon the request of certain shipbuilders and shipowners, have arranged to receive for examination by their surveyors plans and particulars of boilers before the commencement of manufacture, by these means hoping to prevent questions arising after the boilers are finished and on board. This practice has been found to work well in saving time to the surveyors, and in preventing expense, inconvenience, and delay to owners. The senior engineer-surveyor should therefore receive and report on any plans of boilers intended for passenger vessels that may be submitted in due course with the Form Surveys 6. They are not to report on any tracing or plan that is not accompanied by that form. When

[1] These numbers correspond to the clauses of the published Rules.

the surveyor has received plans and tracings of new boilers, or of alterations of boilers, and has approved of them, he will of course be careful in making his examination from time to time to see that they are followed in construction. When he has not had the plans submitted, but is called in to survey a boiler, he will of course measure the parts, note the details of construction, and, if necessary, bore the plates to ascertain their thickness, &c., before he gives his declaration. And in the event of any novelty in construction, or of any departure from the practice of staying and strengthening noted in these Regulations, he should report full particulars to the Board of Trade before fixing the working pressure.

The surveyor cannot declare a boiler to be safe unless he is fully informed as to its construction, material, and workmanship. He should, therefore, be very careful how he ventures to give a declaration for a boiler that he is not called in to survey until after it is completed, and fixed in the ship.

Respecting Stays.—82. In the case of new boilers the surveyor may allow a stress not exceeding 7,000 lbs. per sq. in. of net section on solid iron screwed stays supporting flat surfaces, but the stress should not exceed 5,000 lbs. when the stays have been welded or worked in the fire.

The areas of diagonal stays are found in the following way :—Find the area of a direct stay needed to support the surface, multiply this area by the length of the diagonal stay, and divide the product by the length of a line drawn at right angles to the surface supported to the end of the diagonal stay ; the quotient will be the area of the diagonal stay required.

When gusset stays are used their area should be in excess of that found in the above way.

Girders for Flat Surfaces.—83. When the tops of combustion boxes or other parts of a boiler are supported by solid rectangular girders the following formula should be used for finding the working pressure to be allowed on the girders, assuming that they are not subjected to a greater temperature than the ordinary heat of steam, and in the case of combustion chambers that the ends are fitted to the edges of the tube plate, and the back plate of the combustion box :—

$$\frac{C \times d^2 \times T}{(W-P)\,D \times L} = \text{working pressure.}$$

W = width of combustion box in inches.
P = pitch of supporting bolts in inches.
D = distance between the girders from centre to centre in inches.
L = length of girder in feet.
d = depth of girder in inches.
T = thickness of girder in inches.
C = 500 when the girder is fitted with one supporting bolt.
C = 750 when the girder is fitted with two or three supporting bolts.
C = 850 when the girder is fitted with four supporting bolts.

The working pressure for the supporting bolts and for the plate between them should be determined by the rule for ordinary stays.

Flat Surfaces of Boilers.—84. The pressure on plates forming flat surfaces is found by the following formula :—

$$\frac{C \times (T + 1)^2}{S - 6} = \text{working pressure.}$$

T = thickness of the plate in sixteenths of an inch.
S = surface supported in square inches.
C = constant according to the following circumstances.
C = 100 when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts and washers, the latter being at least three times the diameter of the stay and two-thirds the thickness of the plates they cover.[1]
C = 90 when the plates are not exposed to the impact of heat or flame, and the stays are fitted with nuts only.
C = 60 when the plates are exposed to the impact of heat or flame, and steam in contact with the plates, and the stays fitted with nuts and washers, the latter being at least three times the diameter of the stay and two-thirds the thickness of the plates they cover.
C = 54 when the plates are exposed to the impact of heat or flame, and steam in contact with the plates, and the stays fitted with nuts only.
C = 80 when the plates are exposed to the impact of heat or flame, with water in contact with the plates, and the stays screwed into the plates and fitted with nuts.
C = 60 when the plates are exposed to the impact of heat or flame, with water in contact with the plates, and the stays screwed into the plates, and having the ends riveted over to form substantial heads.
C = 36 when the plates are exposed to the impact of heat or flame, and steam in contact with the plates, with the stays screwed into the plates, and having the ends riveted over to form substantial heads.

In cases where plates are stiffened by T or L irons, and a greater pressure is required for the plates than is allowed by the use of the above constants, the case should be submitted for the consideration of the Board of Trade.

When a circular flat end is bolted or riveted to a cylindrical shell, S in the formula may be taken as the area of the square inscribed in the circle passing through the centres of the bolts or rivets securing the end, provided the angle ring or flange is of sufficient thickness.

When the riveted ends of screwed stays are much worn, or when the nuts are burned, the constants should be reduced, but the surveyor must act according to the circumstances that present themselves at the time of the survey, and it is expected that in cases where the riveted

[1] If the diameter of riveted washers be at least two-thirds the pitch of the stays, and the thickness not less than the plates they cover, the constant may be increased to 150.

When doubling strips are fitted of the same thickness as the plates they cover, and not less in width than two-thirds the pitch of the stays, the constant may be increased to 160.

When doubling plates cover the whole of the flat surface the case should be submitted for the consideration of the Board.

ends of screwed stays in the combustion boxes and furnaces are found in this state it will be often necessary to reduce the constant 60 to about 36.

Compressive Stress on Tube Plates.—85. The surveyors should not in any case allow a greater compressive stress on the tube plates than 8,000 lbs., which is that used in the following formula:—

$$\frac{(D-d)\ T\times 16{,}000}{W\times D}=\text{working pressure.}$$

D = least horizontal distance between centres of tubes in inches.
d = inside diameter of ordinary tubes in inches.
T = thickness of tube plate in inches.
W = extreme width of combustion box in inches from front of tube plate to back of fire box, or distance between combustion box tube plates when boiler is double-ended and the box common to the furnaces at both ends.

Cylindrical Boilers.—87. The Board of Trade consider that boilers well constructed, well designed, and made of good material should have an advantage in the matter of working pressure over boilers inferior in any of the above respects, as unless this is done the superior boiler is placed at a disadvantage, and good workmanship and material will be discouraged. They have therefore caused the following rules to be prepared:—

When cylindrical boilers are made of the best material, with all the rivet holes drilled in place and all the seams fitted with double butt straps, each of at least five-eighths the thickness of the plates they cover, and all the seams at least double-riveted with rivets having an allowance of not more than 75 per cent. over the single shear, and provided that the boilers have been open to inspection during the whole period of construction, then 5 may be used as the factor of safety. The tensile strength of the iron is to be taken as equal to 47,000 lbs. per sq. in. with the grain, and 40,000 lbs. across the grain. But when the above conditions are not complied with, the additions in the following scale should be made to the factor 5, according to the circumstances of each case:—[1]

A†	·15	To be added when all the holes are fair and good in the longitudinal seams, but drilled out of place after bending.
B†	·3	To be added when all the holes are fair and good in the longitudinal seams, but drilled out of place before bending.
C	·3	To be added when all the holes are fair and good in the longitudinal seams, but punched after bending instead of drilled.
D	·5	To be added when all the holes are fair and good in the longitudinal seams, but punched before bending.

[1] If the iron be tested and the elongation measured in a length of 10 ins. is not less than 14 % with and 8 % across the grain, and the surveyors otherwise satisfied as to the quality of the plates and rivets, then 4·5 may be used instead of 5, and the minimum actual tensile strength of the plates used in calculating the working pressure.

E*	·75	To be added when all the holes are not fair and good in the longitudinal seams.
F	·1	To be added if the holes are all fair and good in the circumferential seams, but drilled out of place after bending.
G†	·15	To be added if the holes are fair and good in the circumferential seams, but drilled before bending.
H	·15	To be added if the holes are fair and good in the circumferential seams, but punched after bending.
I†	·2	To be added if the holes are fair and good in the circumferential seams, but punched before bending.
J*	·2	To be added if the holes are not fair and good in the circumferential seams.
K	·2	To be added if double butt straps are not fitted to the longitudinal seams, and the said seams are lap and double riveted.
L	·1	To be added if double butt straps are not fitted to the longitudinal seams, and the said seams are lap and treble riveted.
M	·3	To be added if only single butt straps are fitted to the longitudinal seams, and the said seams are double-riveted.
N	·15	To be added if only single butt straps are fitted to the longitudinal seams and the said seams are treble-riveted.
O	1·0	To be added when any description of joint in the longitudinal seams is single-riveted.
P	·1	To be added if the circumferential seams are fitted with single butt straps and are double-riveted.
Q	·2	To be added if the circumferential seams are fitted with single butt straps and are single-riveted.
R	·1	To be added if the circumferential seams are fitted with double butt straps and are single-riveted.
S‡	·1	To be added if the circumferential seams are lap and are double-riveted.
T	·2	To be added if the circumferential seams are lap and are single-riveted.
U	·25	To be added when the circumferential seams are lap and the strakes of plates are not entirely under or over.
V‡	·3	To be added when the boiler is of such a length as to fire from both ends, or is of unusual length, as in the case of flue boilers, and the circumferential seams fitted as described opposite P, R, and S, but of course when the circumferential seams are as described opposite Q and T, V ·3 will become V ·4.
W*	·4	To be added if the seams are not properly crossed.
X*	·4	To be added when the iron is in any way doubtful, and the surveyor is not satisfied that it is of the best quality.
Y††	1·65	To be added if the boiler is not open to inspection during the whole period of its construction.

Where marked * the allowance may be increased still further if the workmanship or material is very doubtful or very unsatisfactory.

† When the holes are to be rimered or bored out in place, the case should be submitted to the Board as to the reduction or omission of A, B, G, and I as heretofore.

‡ When the middle circumferential seams are double-strapped and double-riveted or lap- and treble-riveted, and the calculated strength not less than 65 % of the solid plate, S ·1 and V ·3 may be omitted. The end circumferential seams in such cases should be at least double-riveted.

†† When surveying boilers that have not been open to inspection during construction, the case should be submitted to the Board as to the factors to be used.

The strength of ordinary joints is found by the following method :—

$$\frac{(\text{Pitch} - \text{diameter of rivet}) \times 100}{\text{pitch}} = \left\{\begin{array}{l}\text{percentage of strength of plate at}\\ \text{joint as compared with the}\\ \text{solid plate.}^1\end{array}\right.$$

$$\frac{(\text{Area of rivet} \times \text{No. of rows of rivets}) \times 100}{\text{pitch} \times \text{thickness of plate}} = \left\{\begin{array}{l}\text{percentage of strength}\\ \text{of rivets as com-}\\ \text{pared with the}\\ \text{solid plate.}^2\end{array}\right.$$

Then take iron as equal to 47,000 lbs.[3] per square inch and use the smaller of the two percentages as the strength of the joint, and adopt the factor of safety as found from the preceding scale :

$$\frac{(47{,}000^3 \times \text{percentage of strength of joint}) \times \text{twice the thickness of the plate in inches}}{\text{Inside diameter of the boiler in inches} \times \text{factor of safety}} = \left\{\begin{array}{l}\text{Pressure to be allowed per square inch on the safety valves. (See the}\\ \text{Formulæ as given in detail in Appendix to the published Rules.)}\end{array}\right.$$

Riveting Butt Straps, &c.—In the case of ordinary zigzag riveting the strength through the plate diagonally between the rivets is equal

[1] *Maximum Pitches for Riveted Joints.*—T = thickness of plate in inches. p = maximum pitch of rivets in inches, provided it does not exceed 10 inches. C = constant applicable from the following table :—

Number of rivets in one pitch	1	2	3	4	5
Constant for lap joints . .	1·31	2·62	3·47	4·14	—
Constant for double butt-strap joints	1·75	3·50	4·63	5·52	6

$$(C \times T) + 1\tfrac{5}{8} = p.$$

When the work is first class, such pitches may be adopted so far as safety is concerned, yet, in some cases, it may be well not to adopt the greatest pitch found by the formula. The maximum pitch should *not*, however, exceed 10 inches with the thickest plates for boiler shells. If in any case the pitch is found to exceed that arrived at by the foregoing formula, for the particular description of joint and thickness of plate, such pitches should *not* be passed, but, in all cases, reported.

[2] If the rivets are exposed to double shear, multiply the percentage as found by 1·75.

[3] See note, p. 300.

to that horizontally between the rivets, when diagonal pitch $= \frac{6}{10}$ horizontal pitch $+ \frac{4}{10}$ diameter of rivet.

Plates that are drilled in place should be taken apart and the burr taken off, and the holes slightly countersunk from the outside.

Butt straps should be cut from plates and not from bars, and should be of as good a quality as the shell plates, and for the longitudinal seams should be cut across the fibre. When the straps are drilled in place they should be taken apart, the burr taken off, and the holes slightly countersunk from the outside.

When single butt straps are used they should be one-eighth thicker than the plates they cover.

The diameter of the rivets should in no case be less than the thickness of the plates of which the shell is made, but it will be found when the plates are thin, or when lap joints or single butt straps are adopted, that the diameter of the rivets should be in excess of the thickness of the plates.

. . . .

Stays for Dished Ends.—Dished ends, unless of the thickness required for a flat end, should be stayed ; but when they are theoretically equal to the pressure needed, when considered as portions of spheres, the stays, when solid, may have a stress of 14,000 lbs. per square inch of net section, but the stress should not exceed 10,000 lbs. when the stays have been welded or worked in the fire, and such stays should be properly distributed. If they are not theoretically equal to the pressure needed they should be stayed as flat surfaces.

Hemispherical End, Manholes, Doors and Domes.—Hemispherical ends subjected to internal pressure may be allowed double the pressure that is suitable for a cylinder of the same diameter and thickness. The ends should not be formed of less than four pieces.

Compensating rings should be fitted around all manholes and openings of at least the same effective sectional area as the plate cut out, and in no case should the rings be less in thickness than the plates to which they are attached. The openings in the shells of cylindrical boilers should have their shorter axes placed longitudinally. It is very desirable that the compensating rings round openings in flat surfaces be made of ∟ or T iron. Cast-iron doors should not be passed.

The neutral part of boiler shells under steam domes should be efficiently stiffened and stayed, as serious accidents have arisen from the want of such precautions.

Hydraulic Test.—The boilers should be tested by hydraulic pressure to twice the working pressure in the presence, and to the satisfaction, of the Board's surveyors.

Circular Furnaces.—88. Circular furnaces with the longitudinal joints welded or made with a butt strap double-riveted, or double butt straps single-riveted :—

$$\frac{90{,}000 \times \text{the square of the thickness of the plate in inches}}{(\text{length in feet} + 1) \times \text{diameter in inches}}$$

= working pressure per square inch, provided it does not exceed that found by the following formula :—

$$\frac{8{,}000 \times \text{thickness in inches}}{\text{diameter in inches}} = \left\{ \begin{array}{l} \text{working pressure} \\ \text{per square inch.} \end{array} \right.$$

The second formula limits the crushing stress on the material to 4,000 lbs. per sq. in.

The length is to be measured between the rings if the furnace is made with rings.

If the longitudinal joints instead of being butted are lap-jointed in the ordinary way and double-riveted, then 75,000 should be used instead of 90,000, but where the lap is bevelled and so made as to give the flues the form of a *true* circle, then 80,000 may be used.

When the material or the workmanship is not of the best quality, the constants given above should be reduced, that is to say, the 90,000 will become 80,000 ; the 80,000 will become 70,000 ; the 70,000 will become 60,000 : when the material and the workmanship are not of the best quality such constants will require to be further reduced, according to circumstances and the judgment of the surveyor, as in the case of old boilers. One of the conditions of best workmanship is that the joints are either double-riveted with single butt straps, or single-riveted with double butt straps, and the holes drilled after the bending is done and when in place, and the plates afterwards taker apart, the burr on the holes taken off, and the holes slightly countersunk from the outside.[1]

The working pressure for corrugated furnaces, practically circular and machine-made, provided the plain parts at the ends do not exceed

[1] The following examples will serve to show the application of the constants for the different cases that may arise :—

Furnaces with Butt Joints and Drilled Rivet Holes.

90,000 where the longitudinal seams are welded.

90,000 where the longitudinal seams are double-riveted and fitted with single butt straps.

80,000 where the longitudinal seams are single-riveted and fitted with single butt straps.

90,000 where the longitudinal seams are single-riveted and fitted with double butt straps.

Furnaces with Butt Joints and Punched Rivet Holes.

85,000 where the longitudinal seams are double-riveted and fitted with single butt straps.

75,000 where the longitudinal seams are single-riveted and fitted with single butt straps.

85,000 where the longitudinal seams are single-riveted and fitted with double butt straps.

Furnaces with Lapped Joints and Drilled Rivet Holes.

80,000	where the	longitudinal	seams are	double-riveted		and bevelled.
75,000	,,	,,	,,	,,	,,	and not bevelled.
70,000	,,	,,	,,	single-riveted		and bevelled.
65,000	,,	,,	,,	,,	,,	and not bevelled.

Furnaces with Lapped Joints and Punched Rivet Holes.

75,000	where the	longitudinal	seams are	double-riveted		and bevelled.
70,000	,,	,,	,,	,,	,,	and not bevelled.
65,000	,,	,,	,,	single-riveted		and bevelled.
60,000	,,	,,	,,	,,	,,	and not bevelled.

In the case of upright fire boxes of donkey or similar boilers, 10 % should be deducted from the constant given above, applicable to the respective classes of work.

6 ins. in length, and the plates are not less than $\frac{5}{16}$ in. thick, should not be greater than found by the following formula :—

$$\frac{9{,}000 \times \text{thickness in inches}}{\text{mean diameter in inches}} = \text{working pressure per square inch.}$$

When the furnaces are riveted in two or more lengths the case should be submitted for consideration, as it may be necessary to make a deduction.

Cylindrical Superheaters.—89. The strength of the joints of cylindrical superheaters and the factor of safety are found in a similar manner as for cylindrical boilers and steam receivers, but instead of using 47,000 lbs. as the tensile strength of iron, 30,000 lbs. is adopted, unless where the heat or flame impinges at, or nearly at, right angles to the plate, then 22,400 lbs. is substituted.

When a superheater is constructed with a tube subject to external pressure, the working pressure should be ascertained by the rules given for circular furnaces, but the constants should be reduced as 30 to 47.

In all cases the internal steam pipes should be so fitted that the steam in flowing to them will pass over all the plates exposed to the impact of heat or flame.

Safety Valve for Superheaters.—Superheaters that can be shut off from the main boilers should be fitted with a parliamentary safety valve of sufficient size, but the least size passed without special written authority should be 3 ins. in diameter.

The flat ends of all boilers, as far as the steam space extends, and the ends of superheaters, should be fitted with shield, or baffle, plates where exposed to the hot gases in the uptake, as all plates subject to the direct impact of heat or flame are liable to get injured unless covered with water.

Haystack Boilers.—90. As the uptakes of haystack boilers and others of this type are especially liable to injury from overheating unless careful precautions are taken while steam is being raised, the surveyor should in all cases endeavour to persuade makers and owners to make the strength of the uptakes considerably in excess of that required for ordinary superheaters subject to external pressure.

The employment of bowling rings is beneficial by adding to the strength as well as allowing for expansion, but if there is a difficulty in getting these fitted, hoops riveted round, although not so desirable as bowling rings, can be employed to increase the resistance of the tubes against collapse. The use of bowling rings with a moderate thickness of plate is better than very thick plating. This applies to the uptakes of all boilers of this type, including ordinary vertical donkey boilers. Bowling rings fitted to all such uptakes would be a decided advantage in allowing for expansion. When flaming coal is used extra care is required, and extra strength absolutely necessary.

. . . .

Steel Boilers.

92. The following should guide the Board's surveyors when the general quality of the steel has been found suitable for marine boilers.

Makers' Tests.—The steel makers or boiler makers should test one or more strips or pieces cut from *each* plate and bar for tensile strength and elongation, and stamp both results on each plate or bar. When practicable the plates or bars should be so stamped that the marks can be easily seen when the boiler is constructed.

Surveyor's Tests of Plates.—93. The surveyor is not obliged to witness the foregoing tests, although it is very desirable that he should when his other duties will allow him to do so, but he should see that all the plates and bars are properly marked. He should, however, select of each thickness *at least* one in four of the plates to be used in the construction of the boiler, either at the steel works or at the boiler makers' works, and witness the testing of at least one strip or piece cut from each selected plate; but when shell plates exceed 15 ft. in length, there should be a tensile test from each end, and when they exceed 20 ft. in length and at the same time exceed 6 ft. in breadth, or exceed $2\frac{1}{2}$ tons in weight, there should be a tensile test from each corner. In the latter cases the testing of each plate should be witnessed by the surveyor. The mean of the results of the tests, if the latter fall within the Board's requirement as stated below, should be stamped on the plates. If a large number of failures take place in the 25 % selected, the surveyor should see more than 25 % of the plates to be used in the boiler satisfactorily tested.

Tensile and Bending Tests.—If for the plates from which the surveyor selects the above proportion a greater stress is wished than is allowed for iron, tests for tensile strength and elongation should be made, also a few tempering and bending tests, and those for which no reduction of thickness is asked may be tested for resistance to bending and tempering only, if preferred. In the latter case the tensile strength and elongation stamped on each plate should be reported by the surveyor to the Board of Trade, along with the results of the bending and tempering tests. From the plates and bars, the tests of which have been stated to have been made by the steel maker, and not witnessed by the surveyor, the surveyor may, if he thinks it advisable, select any plates or bars after they are in the boiler yard and require specimens to be cut off and tested. If the results are not satisfactory the whole of the plates, except those which were tested and found satisfactory by the surveyor, may be liable to be rejected.

Test Strips.—94. The breadth of test strips for tensile stress should be about 2 ins., and the elongation taken in a length of 10 ins. should be about 25 %, and not less than 18 % when tested in the normal condition, in which condition the Board prefer the tests to be made; but if the plates are annealed, that is, heated to a red heat in a *plate* furnace, and immediately they are at that heat taken out and placed on the mill floor to cool, the elongation should not be less than 20 %. The test pieces must not be annealed after they are cut off the plates. When the plates are not taken out of the furnace immediately they are red hot, or if allowed to cool down in the furnace, or are covered with ashes or other non-conducting substance, it should be reported to the Board for their consideration and decision. The surveyor should always report to the Board whether the plates have been annealed, or if in the normal condition when the test pieces were cut off. The test strips must be carefully prepared and measured,

and they should be cut from the plate by a planing or shaping machine. The skin of the test pieces should not be removed by planing, shaping, or otherwise, the edges only being planed or shaped, and in no case should test pieces be prepared or reduced in size by hammering or otherwise working on the anvil.

Bending Tests.—95. The bending test for plates *not* exposed to flame should be made with strips in the same condition as the plates, and occasionally also some tempering tests should be made. Strips cut from furnaces, combustion boxes, &c., should be heated to a cherry-red, then plunged into water at about 80°, and kept there until of the same temperature as the water, and then bent. The bending and tempering strips should not be less than 2 ins. broad and 10 ins. long, and they should be bent until they break, or until the sides are parallel at a distance from each other of not more than (3) three times the thickness of the plate.

Tensile Strength, &c.—96. When full allowance over iron is wished, the tensile strength of the plates *not* exposed to flame should be not less than 27 tons, and should not exceed 32 tons, per sq. in. of section, and 27 tons should be the stress used in the calculations for cylindrical shells if the plates comply with all the conditions as stated herein, but for each ton the minimum tensile strength of the plate is above 27 tons, 1 ton may be added to the 27, provided the Surveyor witnesses the testing of all the plates. The tensile strength of furnace, flanging, and combustion box plates, may range from 26 to 30 tons per sq. in.

Surveyors' Tests, Stay and Rivet Bars.—97. Stay and rivet bars should be tested for tensile strength and elongation, viz. : one bar in 20 when the diameter of the bar does not exceed 1 in. ; one bar in 12 when not over 1½ in. ; and one bar in 8 when the diameter exceeds 1½ in.

Tensile Strength of Stays, &c.—The tensile strength of stay bars should be from 27 to 32 tons per sq. in., with an elongation of about 25 % and not less than 20 % in a length of 10 ins. Solid steel screwed stays which have not been welded or otherwise worked after heating may be allowed a working stress of 9,000 lbs. per sq. in. of nett section, provided the tensile strength and elongation are as stated. Steel stays which have been welded or worked in the fire have been found to be unreliable, therefore they should not be passed.

Rivet Bars and Rivets.—The tensile strength of rivet bars should be from 26 to 30 tons per sq. in. with an elongation of not less than 25 per cent. in a length of 10 inches. The rivets before being tested should be carefully prepared, and the elongation should when practicable be taken in a length of 2½ times the diameter of the prepared part. The tensile strength of the rivets should be from 27 to 32 tons per sq. in., and the contraction of area about 60 %.

If the original size of the bars for rivets or stays be reduced before testing it must be done in the lathe or by machine : test pieces of any kind should not be prepared by heating and drawing down.

Perforating and Annealing.—98. The rivet holes in the furnace and shell seams should be *drilled*, but if it is wished to punch them and afterwards bore or anneal the plates in a proper furnace, the particulars of the punching and boring or annealing should be submitted

to the Board of Trade for consideration before being done, but all punched holes should be made after bending.

In all cases where assent has been given for plates to be punched after bending and then annealed, the makers should stamp the plates with the words 'punched after bending and then annealed,' and in all cases where assent has been given for punching and afterwards boring plates the words 'punched and then bored' should be stamped on the plates.

Constants for Flat Surfaces, Girders, and Plain Furnaces.—99. If the flanging plates and those exposed to flame comply with the foregoing conditions, the constants in the Board's rules for iron boilers may be increased as follows :—

The constants for flat surfaces when they are supported by stays screwed into the plate and riveted, 10 %.

The constants for flat surfaces when they are supported by stays screwed into the plate and nutted, or when the stays are nutted in the steam space, 10 %. This is also applicable to the constants for flat surfaces stiffened by riveted washers or doubling strips, and supported by nutted stays.

The constants for combustion box girders, 10 %.

The constants for plain furnaces, 10 %.

Furnaces Corrugated or Ribbed and Grooved.—100. When the furnaces are corrugated and machine made, such as made by Messrs. The Leeds Forge Company, or ribbed and grooved as made by Messrs. J. Brown and Company, Sheffield, if they are practically true circles, provided the plain parts at the ends do not exceed 6 ins. in length, and the plates are not less than $\frac{5}{16}$ in. thick, the working pressure is found by the following formula :—

$$\frac{14{,}000 \times T}{D} = \text{working pressure.}$$

T = thickness in inches.

D = outside diameter in inches, measured at the bottom of the corrugations when the furnace is corrugated, or over the plain parts when it is ribbed and grooved.

In corrugated furnaces the pitch of the corrugations should not exceed 6 ins., and the depth from top of corrugation outside to bottom of corrugation inside should not be less than 2 ins., and the plates at the ends not unduly thinned in the flanging. The plates of ribbed and grooved furnaces should be formed by rolling, the ribs should not be less than $1\frac{3}{16}$ ins. above the plain parts, the depth of the grooves not more than $\frac{3}{4}$ in., the length between the centres of the ribs not over 9 ins., and the ends rolled slightly thicker than the plain parts, and not reduced at any part by flanging, &c., below the thickness of the body of the furnace.

If the furnace is riveted in one or more lengths the case should be submitted for consideration.

Furnaces made up of Flanged Rings.—101. When horizontal furnaces of ordinary diameter are constructed of a series of rings welded longitudinally, and the ends of each ring flanged and the rings riveted together, and so forming the furnace, the working pressure is found by the following formula, provided the length in inches between the

centres of the flanges of the rings is not greater than (135 T − 12), and that the conditions which follow the formula are complied with :—

$$\frac{8{,}800 \times T}{3 \times D}\left(5 - \frac{l + 12}{67 \cdot 5 \times T}\right) = \text{working pressure.}$$

T = thickness of plate in inches.
l = length between centre of flanges in inches.
D = outside diameter of furnace in inches.

The radii of the flanges on the fire side should be about $1\frac{1}{2}$ ins. The depth of the flanges from the fire side should be three times the diameter of the rivet plus $1\frac{1}{2}$ ins., and the thickness of the flanges should be as near the thickness of the body of the plate as practicable. The distance from the edge of the rivet holes to the edge of the flange should not be less than the diameter of the rivet, and the diameter of the rivets at least $\frac{3}{8}$ in. greater than the thickness of the plate. The depth of the ring between the flanges should be not less than three times the diameter of the rivets, the fire edge of the ring should be about the termination of the curve of flange, and the thickness not less than half the thickness of the furnace plate. Turned rings are such as may be considered a first-class method of construction.

The holes in the flanges and rings should be drilled in place if practicable, but if not drilled in place they should be drilled sufficiently small, and afterwards when in place rimered out until the holes are quite fair ; the holes should be slightly tapered and the heads of the rivets of moderate size.

After all welding, and after all flanging and heating, each ring should be efficiently annealed in one operation.

Compressive Stress on Tube Plates.—102. A greater compressive stress should not be allowed on tube plates than 10,000 lbs., which is that used in the following formula :—

$$\frac{(D-d)\ T \times 20{,}000}{W \times D} = \text{working pressure.}$$

D = least horizontal distance between centres of tubes in inches.
d = inside diameter of ordinary tubes in inches.
T = thickness of tube plate in inches.
W = extreme width of combustion box in inches from front of tube plate to back of fire box, or distance between combustion-box tube plate plates when boiler is double-ended and the box common to the furnaces at both ends.

Plate and Rivet Section.—103. When full allowance is wished, the rivet section, if iron, in the longitudinal seams of cylindrical shells should, when those seams are lapped, be at least $\frac{13}{8}$ times the net plate section, and if steel rivets are used their section should be at least $\frac{28}{23}$ of the net section of the plate if the tensile stress of the rivets is not less than 27 tons and not more than 32 tons per sq. in. In calculating the working pressure, the percentage strength of the rivets may be found in the usual way by the Board's rules, but in the case of iron rivets the percentages found should be divided by $\frac{13}{8}$, and in the case of steel rivets by $\frac{28}{23}$, the results being the percentages required. If the per-

centage strength of the rivets by calculation is less than the calculated percentage strength of the plate, calculate the working pressure by both percentages. When using the percentage strength of the plate, 4·5 plus the additions suitable for the method of construction as by the Board's rules for iron boilers may be used as the nominal factor of safety, but when using the percentage strength of the rivets 4·5 may be used as the factor of safety. The less of the two pressures so found is the working pressure to be allowed for the cylindrical portion of the shell, or otherwise in accordance with the formulæ in Appendix.

Local Heating to be avoided.—104. Local heating of the plates should be avoided, as many plates have failed from having been so treated.

Annealing.—All plates that are punched, flanged, or locally heated must be carefully annealed after being so treated.

Welding, &c.—Steel plates which have been welded should not be passed if subject to a tensile stress, and those welded and subject to a compressive stress should be efficiently annealed.

In other respects the boilers should comply with the rules for iron boilers.

If the tests are to be made at the steel works, the boiler makers should inform the surveyors in their district when and where they will be made, so that a surveyor from the nearest district to the steel works may be instructed to attend to them. The surveyors should have due notice—two or three days when the plates &c. will be ready for the test pieces to be cut from them. As soon as possible after tests are made, the results should be submitted for the Board's consideration. (Surveyors should report all cases of failures of steel plates &c. which may come to their knowledge.)

Steel for Superheaters, &c.—105. If steel is proposed to be used in superheaters the particulars should be submitted to the Board of Trade for consideration, but in all cases it should be discouraged for this purpose. This applies to the unshielded uptakes of all boilers, including ordinary vertical donkey boilers.

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Boiler Tracing, &c.—107. Difficulty has been experienced with regard to the survey of steel boilers owing to the fact that some makers were not aware, at the time the boilers were commenced, that a Board of Trade certificate would be necessary, and the makers have therefore omitted to submit tracings until the boilers have been nearly completed. Tracings of boilers may therefore be received for examination upon payment of the usual fee of 2*l*., and the surveyors may proceed as far as witnessing the hydraulic test before any further instalment of the survey fee is paid. Engineers and boiler makers should be advised of this arrangement.

Boilers.—General.

Donkey Boilers.—108. Donkey boilers that are in any way attached to, or connected with, the main boilers, or with the machinery used for propelling the vessel, should be surveyed and be fitted the same way as the main boilers, and have a water and steam gauge, and all other fittings complete, and, as regards safety valves, should comply with the same regulations as the main boilers.

Launch Boilers.—109. The boilers of steam launches forming part of the statutory boat capacity of passenger steamers should, as regards construction, strength, material, safety valves, and other fittings, comply with the same regulations as the main boilers.

Stop Valves.—110. No boiler or steam chamber should be so constructed, fitted, or arranged that the escape of steam from it through the safety valves required by the Act of Parliament can be wholly, or partially, intercepted by the action of another valve.

A stop valve should always be fitted between the boiler and the steam pipe, and, where two or more boilers are connected with a steam receiver or superheater, between each boiler and the superheater or steam receiver. The object of this is obvious, viz. to avoid the failure of all the boilers through the failure of one. The necks of stop valves should be as short as practicable.

Declarations should not be given for new steamers, for steamers that have not had passenger certificates before February 1871, nor for steamers that have had new boilers since February 1871, unless stop valves are fitted between the boilers and the steam pipes.

Water-gauges, Test Cocks, &c.—111. Each boiler should be fitted with a glass water-gauge, at least three test cocks, and a steam gauge. Boilers that are fired from both ends, and those of unusual width, should have a glass water gauge and three test cocks at each end or side, as the case may be. When a steamer has more than one boiler, each boiler should be treated as a separate one, and have all the requisite fittings.

When the upper and lower glass water-gauge cocks are not attached directly to the shell of the boiler, but a standpipe or column is fitted, cocks or valves, although the former are more desirable, should as a general rule be fitted between the boiler and the standpipes &c. Cocks or valves need not, however, be insisted on in cases where the columns, standpipes, &c., are of moderate length and of extra strength, provided that the diameter of the bore at any part is not less than 3 ins.

If the column, standpipes, &c., are of less diameter than 3 ins., and the pipes are bolted to the boiler without the intervention of cocks or valves, the arrangement need not be objected to, if otherwise satisfactory, providing there is no difficulty in keeping the passages at the ends clear, and ascertaining that they are so. To do this it will be necessary that the passage in the part of the column between the top and bottom gauge-glass cocks be cut off or closed, which may be done permanently, or by the interposition of a cock or valve at that part. The latter is a convenient and desirable arrangement even when cocks are fitted on the boiler.

In the case of high pressures, it is desirable that the cocks or valves which prevent the escape of water or steam from the boiler be fitted with handles which can be expeditiously manipulated from a convenient position.

It is desirable in all cases that test cocks be fitted directly to the skin of the boiler, and when the passage in the part of the column to which the glass water-gauge cocks are attached (if so fitted) is permanently cut off, the test cocks *must* be fitted directly to the skin of the boiler.

The surveyors should satisfy themselves by actual examination whether the glass water-gauges of the boilers of the vessels they survey are fitted with automatic valves or fittings, as the existence of such fittings cannot always be ascertained by external examination. In all cases full particulars of automatic gauges should be submitted for consideration before the gauges are passed.

No new arrangement to be sanctioned until plans have been submitted to Board of Trade.—112. Surveyors are to be most careful not to give any official sanction to any new arrangement or construction of marine steam boilers, including boilers with curved ends, without first obtaining the permission of the Board in writing, nor are they allowed to give any written approval of any invention, or arrangement, unless by direction of the Board of Trade; and whenever they know that any invention or new arrangement is to be fitted to a vessel that is intended to have a passenger certificate, they should, as soon as possible, with a view to preventing subsequent delays and questions, obtain plans and submit the same for the consideration of the Board of Trade.

When any deviation from an approved plan is made, it should be submitted for the Board's consideration, and when any deviation is sanctioned it is only for that particular case, unless otherwise stated. Surveyors should in all cases record on their declarations whether the boilers are made of iron or steel, and if made partly of steel and partly of iron they should specify for what parts either metal is used.

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Hydraulic Test.—115. Surveyors should see all new boilers, and boilers that have been taken out of the ship for a thorough repair, tested by hydraulic pressure up to at least double the working pressure that will be allowed, previous to the boilers being placed in the vessel, and before they are lagged, to test the workmanship, &c.; but the working pressure is to be determined by the strength of stays, thickness of plates, and strength of riveting, &c., and not by the hydraulic test.

The hydraulic test should not be witnessed by the surveyors in any case where the Board's regulations as to strength, material, method of construction, treatment, &c., are not complied with, unless they have previously submitted the details of the particular case for the consideration of the Board and obtained authority to witness the test.

The hydraulic test should in no case exceed twice the calculated working pressure of the boiler, and it is never to be applied until the boiler has been examined in accordance with clause 114, and until the strength has been calculated from the necessary measurements taken from the boiler itself.

When the boilers are in the vessel the surveyor may, at any time he thinks it necessary, before he gives a declaration, have them tested by hydraulic pressure to satisfy himself as to any doubtful part, or of places not easy of access, care being taken in the case of old boilers not to overstrain them; but the test must always exceed the working pressure.

The hydraulic test should be applied to the boilers of all steamers that have not previously had a passenger certificate, before a declaration is granted for them.

When a vessel is partially surveyed by one surveyor, and the survey is completed and a declaration granted by another, if the surveyor who witnesses the test of the boilers by hydraulic pressure has an opportunity of examining them inside and outside after the test, such surveyor should determine the pressure to be allowed on the boilers in question, taking care to inform the makers, owners, or agents, and the surveyor who is ultimately to grant a declaration, what pressure should in his opinion be allowed on them.

The amount of the hydraulic pressure test, and the date on which it was last applied to the boilers, should be inserted in the surveyor's declaration, and recorded in the Office Boiler Book.

Examination and Testing of Steam Pipes.—116. Surveyors should pay particular attention to the examination and testing of steam pipes.

All new copper steam pipes should be tested by hydraulic pressure to not less than twice and not more than two and one-half times the working pressure, unless the case has been specially submitted to the Board for consideration.

Wrought-iron lap-welded steam pipes should be tested by hydraulic pressure when new to at least twice the working pressure, but not to more than three times, unless the case has been specially submitted to the Board for consideration.

As regards old pipes the surveyor may at any time he thinks it necessary, before he gives a declaration, have them tested by hydraulic pressure to satisfy himself as to any doubtful part, but they should be tested, with the lagging removed for examination, at least once in about every four years, to not less than double the working pressure. A record of the test should be kept in the Office Boiler Book.

There should be efficient means provided for draining all the steam pipes.

Copper Pipes.—117. The working pressure of well-made copper pipes when the joints are brazed is found by the following formula :—

$$\frac{6000 \times (T - \frac{1}{16})}{D} = \text{working pressure.}$$

T = thickness in inches.
D = inside diameter in inches.

When the pipes are solid drawn and not over 8 ins. diameter, substitute in the foregoing formula $\frac{1}{32}$ for $\frac{1}{16}$.

Wrought-Iron Pipes.—118. The internal pressure on wrought-iron pipes made of good material, which are lap-welded and are a sound job, may be determined by the following formula, provided that the thickness is not less than $\frac{1}{4}$ in. :—

T = thickness in inches.
D = diameter inside in inches.

$$\frac{6000 \times T}{D} = \text{working pressure.}$$

Feed Pipes.—119. Feed pipes should be made sufficient for a pressure 20 % in excess of the boiler pressure.

Expansion Joints.—120. In all cases in which a socket expansion joint is fitted to a bent steam pipe, the surveyor should require a

fixed gland and bolts to be fitted, in order to prevent the end of the pipe being forced out of the socket. This regulation should be complied with in all cases of bent pipes fitted with socket expansion joints. It is also desirable that fixed glands and bolts should be fitted to the expansion joints of straight steam pipes, as cases have occurred, particularly with small straight pipes, in which the ends of the pipes have been forced out of the sockets.

Cast Iron in Boilers and Steam Pipes, Stand Pipes, Cocks, &c.—121. In all boilers in which the surveyors find that cast iron is employed in such a manner as to be subject to the pressure of steam or water, they are directed to report the circumstances to the Board of Trade, in order that they may receive instructions how to act. Cast-iron standpipes or cocks through which hot brine would have to pass should never be passed. Cast iron should not be used for stays, and surveyors should also discourage the use of cast iron for chocks and saddles for boilers. Particular attention should be paid to the chocking of boilers, more especially when they are fired athwartships.

Pressure fixed by one Surveyor not to be increased by another.—122. A pressure once allowed on the boiler of a passenger steam ship is not, *under any circumstances whatever*, to be increased unless the surveyor has previously referred the matter to the Board. In cases where a surveyor is of opinion that an increased pressure may with safety be allowed, he should communicate with the surveyor who last surveyed the vessel; and if, on learning the reasons why the existing pressure was formerly allowed, the surveyor is still of opinion that it may be increased, he should communicate all the facts of the case to the Board of Trade; but, as above stated, the pressure should not in any case be increased until the question has been decided by them.

SAFETY VALVES.

Provisions as regards Safety Valves.—123. The engineer-surveyor shall declare, amongst other things, the limits of the weight to be placed on the safety valves; that the safety valves are such and in such condition as required by the Act, and that the machinery is sufficient for the service for the time he fixes, and is in good condition for that time.

The locked-up valves, *i.e.* those out of the control of the engineer when steam is up, should have an area not less, and a pressure not greater, than those which are not locked up, if any such valves are fitted.

Cases have come under the notice of the Board of Trade in which steam ships have been surveyed, and passed by the surveyors, with pipes between the boilers and the safety-valve chests. Such arrangement is not in accordance with the Act, which distinctly provides that the safety valves shall be upon the boilers.

The surveyors are instructed that in all *new boilers*, and whenever *alterations can be easily made*, the valve chest should be placed directly on the boiler; and the neck, or part between the chest and the flange which is bolted on to the boiler, should be as short as possible and be cast in one with the chest.

The surveyors should note that it is not intended by this instruction

that vessels with old boilers which have been previously passed with such an arrangement should be detained for the alterations to be carried out.

Of course in any case in which a surveyor is of opinion that it is positively dangerous to have a length of pipe between the boilers and the safety-valve chest, it is his duty at once to insist on the requisite alterations being made before granting a declaration.

If any person place an undue weight on the safety valve of *any* steam ship, or, in the case of steam ships surveyed under the Act, increase such weight beyond the limits fixed by the engineer-surveyor, he shall, in addition to any other liabilities he may incur by so doing, incur a penalty not exceeding one hundred pounds.

Area of Safety Valves.—124. The area per square foot of fire-grate surface of the locked-up safety valves should not be less than that given in the following tables opposite the boiler pressure intended, but in no case should the valves be less than two inches in diameter. This applies to new vessels or vessels which have not received a passenger certificate.

When, however, the valves are of the common description, and are made in accordance with the tables, it will be necessary to fit them with springs having great elasticity, or to provide other means to keep the accumulation within moderate limits; and as boilers with forced draught may require valves considerably larger than those found by the tables, the design of the valves proposed for such boilers, together with the estimated coal consumption per square foot of fire-grate, should be submitted to the Board for consideration.

In ascertaining the fire-grate area, the length of the grate should be measured from the inner edge of the dead plate to the front of the bridge, and the width from side to side of the furnace on the top of the bars at the middle of their length.

In the case of vessels that have not had a passenger certificate, if there is only one safety valve on any boiler, the surveyor should not grant a declaration without first referring the case to the Board for special instructions.

Safety-Valve Areas.

Boiler Pressure	Area of Valve per Square Foot of Fire Grate	Boiler Pressure	Area of Valve per Square Foot of Fire Grate	Boiler Pressure	Area of Valve per Square Foot of Fire Grate
15	1·250	30	·833	45	·625
16	1·209	31	·815	46	·614
17	1·171	32	·797	47	·604
18	1·136	33	·781	48	·595
19	1·102	34	·765	49	·585
20	1·071	35	·750	50	·576
21	1·041	36	·735	51	·568
22	1·013	37	·721	52	·559
23	·986	38	·707	53	·551
24	·961	39	·694	54	·543
25	·937	40	·681	55	·535
26	·914	41	·669	56	·528
27	·892	42	·657	57	·520
28	·872	43	·646	58	·513
29	·852	44	·635	59	·506

Safety-Valve Areas—cont.

Boiler Pressure	Area of Valve per Square Foot of Fire Grate	Boiler Pressure	Area of Valve per Square Foot of Fire Grate	Boiler Pressure	Area of Valve per Square Foot of Fire Grate
60	·500	107	·307	154	·221
61	·493	108	·304	155	·220
62	·487	109	·302	156	·219
63	·480	110	·300	157	·218
64	·474	111	·297	158	·216
65	·468	112	·295	159	·215
66	·462	113	·292	160	·214
67	·457	114	·290	161	·213
68	·451	115	·288	162	·211
69	·446	116	·286	163	·210
70	·441	117	·284	164	·209
71	·436	118	·281	165	·208
72	·431	119	·279	166	·207
73	·426	120	·277	167	·206
74	·421	121	·275	168	·204
75	·416	122	·273	169	·203
76	·412	123	·271	170	·202
77	·407	124	·269	171	·201
78	·403	125	·267	172	·200
79	·398	126	·265	173	·199
80	·394	127	·264	174	·198
81	·390	128	·262	175	·197
82	·386	129	·260	176	·196
83	·382	130	·258	177	·195
84	·378	131	·256	178	·194
85	·375	132	·255	179	·193
86	·371	133	·253	180	·192
87	·367	134	.251	181	·191
88	·364	135	·250	182	·190
89	·360	136	·248	183	·189
90	·357	137	·246	184	·188
91	·353	138	·245	185	·187
92	·350	139	·243	186	·186
93	·347	140	·241	187	·185
94	·344	141	·240	188	·184
95	·340	142	·238	189	·183
96	·337	143	·237	190	·182
97	·334	144	·235	191	·181
98	·331	145	·234	192	·181
99	·328	146	·232	193	·180
100	·326	147	·231	194	·179
101	·323	148	·230	195	·178
102	·320	149	·228	196	·177
103	·317	150	·227	197	·176
104	·315	151	·225	198	·176
105	·312	152	·224	199	·175
106	·309	153	·223	200	·174

Examination of Safety Valves.—125. The surveyor, in his examination of the machinery and boilers, is particularly to direct his attention to the safety valves, and whenever he considers it necessary, he is to satisfy himself as to the pressure on the boiler by actual trial.

The surveyor is to fix the limits of the weight to be placed on the safety valves, and the responsibility of issuing a declaration before he

is fully satisfied on the point is very grave. The law places on the surveyor the responsibility of 'declaring' that the boilers are in his judgment sufficient with the weights he states.

The surveyor is to examine the whole of the valves, weights, and springs at every survey.

The responsibility of seeing to the efficiency of the mode by which the valves are fitted so as to be out of the control of the engineer when steam is up rests with the surveyor, as long as it is efficient, and the method adopted is approved of by the Board of Trade.

The safety valves should be fitted with lifting gear, so arranged that the two or more valves on any one boiler can at all times be eased together, without interfering with the valves on any other boiler. The lifting gear should in all cases be arranged so that it can be worked by hand either from the engine-room or stoke-hole.

Care should be taken that the safety valves have a lift equal to at least one-fourth their diameter; that the openings for the passage of steam to and from the valves, including the waste-steam pipe, should each have an area not less than the area of valves required by clause 124; and that each valve box has a drain pipe fitted at its lowest part. In the case of lever-valves, if the lever is not bushed with brass, the pin must be of brass; iron and iron working together must not be passed. Too much care cannot be devoted to seeing that there is proper lift, and free means of escape of waste steam, as it is obvious that unless the lift and means for escape of waste steam are ample, the effect is the same as reducing the area of the valves or putting on an extra load. The valve seats should be secured by studs and nuts.

The surveyors are, as far as in their power, to make the opinion of the Board on these points generally known to the owners of passenger steamers.

Surveyor to see Valves Weighted.—126. When the surveyor has determined the amount of pressure, he is to see the valves weighted accordingly, and the weights or springs fixed in such a manner as to preclude the possibility of their shifting or in any way increasing the pressure. The limits of the weight on the valves is to be inserted in the declaration, and should it any time come to a surveyor's knowledge that the weights or the loading of the valves have been shifted, or otherwise altered, or that the valves have been in any way interfered with, so as to increase the pressure, without the sanction of the Board of Trade, he is at once to report the facts to the Board of Trade.

Spring Safety Valves.—127. If the following conditions are complied with, the surveyor need raise no question as to the substitution of spring-loaded valves for dead-weighted valves :—

1. That at least two valves are fitted to each boiler.

2. That the valves are of the proper size, as by clause 124.

3. That the springs and valves are so cased in that they cannot be tampered with.

4. That provision is made to prevent the valves flying off in case of the springs breaking.

5. That the requisite safety-valve area is cased in and locked up in the usual manner of the Government valves.

6. That screw lifting gear is provided to ease all the valves, as by clause 125.

7. That the size of the steel of which the springs are made is in accordance with that found by the following formula :—

$$\sqrt[3]{\frac{s \times D}{c}} = d ;$$

s = the load on the spring in lbs.
D = the diameter of the spring (from centre to centre of wire) in inches.
d = the diameter, or side of square, of the wire in inches.
c = 8,000 for round steel.
c = 11,000 for square steel.

8. That the springs are protected from the steam and impurities issuing from the valves.

9. That when valves are loaded by direct springs, the compressing screws abut against metal stops or washers, when the loads sanctioned by the surveyor are on the valves.

10. That the springs have a sufficient number of coils to allow a compression under the working load of at least one-quarter the diameter of the valve.

Spring-loaded Valves to be Tested under Steam.—128. In no case is the surveyor to give a declaration for spring-loaded valves unless he has examined them and is acquainted with the details of their contruction, and unless he has tried them under full steam, and full firing, for at least twenty minutes with the feed-water shut off and stop valve closed, and is fully satisfied with the result of the test. In special cases, or when the valves are of novel design, the results of the tests under full steam should be reported to the Board, but if the surveyors are fully satisfied with the results of the tests they need not delay the granting of the declaration for the vessel subject to approval of the Board. If the accumulation of pressure exceed 10 % of the loaded pressure, he should not give his declaration without first reporting the case to the Board of Trade, accompanied by a sketch, and full particulars of the trial and the strength pressure of the boilers.

Plans of New Designs or of Alterations in Details to be submitted.—129. In the case of valves of which the principle and details have already been passed by the Board of Trade, the surveyor need not require plans to be submitted so long as the details are unaltered, of which he must fully satisfy himself; but in any new arrangement of valves, or in any case in which any detail of approved valves is altered, he should, before assuming the responsibility of passing them, report particulars, with a drawing to scale, to the Board of Trade. He can make this drawing himself from the actual parts of the valves fitted, but in order to save time, and to facilitate the survey, the owners or makers of engines may prefer to send in tracings of their own, before the valves are placed on the boiler. If they do this the survey can be more readily made, and delay and expense may be saved to owners, as the surveyor will not then have to spend his time, and delay the ship, in preparing drawings and comparing them with the valves.

The tracings of new safety-valve designs should, if possible, be

transmitted to the Board of Trade for consideration before the construction of the safety valves is commenced.

In some spring valves the accumulation of pressure has reached cent. per cent., and therefore if the surveyor had not required a trial, he would have passed valves which would have caused a pressure on the boiler double that intended by him. And in some cases in which the increase of pressure has not been great, defects that would have rendered the valves highly dangerous have been discovered on an examination of drawings.

The surveyors should arrange with manufacturers so that the surveyors may have the designs of valves which the manufacturers intend to use. An easy method of facilitating this matter is for the manufacturer to leave in the local surveyor's office a plan or plans of his valve or valves when once agreed to, and then afterwards to inform the surveyor that the valves fitted are according to drawing A, B, or C, as the case may be. By this means, when once a design has been agreed upon, and is adhered to, all subsequent questions and delays will be prevented.

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BOILERS AND MACHINERY, &c.

What Machinery is to be Surveyed.—133. The machinery to be surveyed comprises the engines and boilers used for propelling the vessel, and all the machinery connected therewith. The boilers of donkey engines are to be surveyed with the boilers and machinery of the vessel, when they are in any way connected with them, thus:—even when the donkey boiler is connected with the donkey engine that is used for pumping the water into the main boiler, it is considered to be connected to the machinery and boilers used for propelling the ship. But boilers and machinery used for loading or unloading the vessel, or used exclusively for purposes unconnected with the motive power of the vessel, do not form a part of the machinery required to be surveyed by the Merchant Shipping Act of 1854.

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Feed Check Valves and Cocks. Separate Feed Arrangements.—136. Each boiler of a passenger vessel, whether old or new, should be fitted with suitable check valves between it and the feed pipes, and the boilers of all *new* passenger vessels, passenger tugs, and other small passenger vessels, should be fitted with separate feeding arrangements in addition to, but unconnected with, the main feed pipes and valves. It is desirable that the main feed check valve chest on each boiler be separate and distinct from that of the auxiliary feed, and that a stop cock or stop valve be fitted in each chest or between each chest and the boiler, so that the latter may be shut off, and either of the check valves examined while the other feed is at work. In very small vessels an efficient hand-pump, instead of the donkey pump, may be passed if the surveyor has satisfied himself as to its efficiency when steam is up, and provided there are separate feed pipes and valves, as directed above. This is to apply also in the case of old vessels of the above description, when being fitted with new boilers.

The surveyor should discourage the practice of using the same pump for the bilges and for feeding boilers.

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Guards on Blow-off Cocks.—138. With a view to the prevention of accidents to boilers through the blow-off cocks being left open after the boiler is run up, and to prevent water getting accidentally or intentionally into the ship by cocks being left open, all blow-off cocks and sea connections below the plates or out of sight should be fitted with a guard over the plug, with a featherway in the same, and a key on the spanner, so that the spanner cannot be taken out unless the plug or cock is closed. When cocks are in sight, guards need not be fitted provided the spanners are secured to the plugs by pins. The spanners should not be shrunk on the heads of the plugs. One cock should be fitted to the boiler, and another cock on the skin of the ship or on the side of the Kingston valve.

Non-return Valves to Pipes.—139. In all cases where pipes are so led or placed that water can run from the boiler or the sea into the bilge, either by accidentally or intentionally leaving a cock or valve open, they should be fitted with a non-return valve and a screw not attached, but which will set the valve down in its seat when necessary; the only exception to this is the fireman's ash cock, which must have a cock or valve on the ship's side, and be above the stoke-hole plates.

. . . .

Tests of Material and Fees.—142. In the case of boilers and machinery, it may be impracticable to apply any satisfactory tests to the material used after the boilers or machinery are placed in the ship, without great inconvenience and delay to the shipowners.

With a view to obviate such inconvenience and delay, the surveyor is authorised, at the request of the mannfacturer or of the person for whose intended ship the material is being manufactured, to inspect and test such material during manufacture.

Such request must be made on Form Survey 6, or on Form Survey 6A, or on such other form as the Board of Trade from time to time direct, and must be accompanied with a written undertaking to pay to the Board of Trade on the delivery of the certificate, (1) such sum for travelling expenses as the Board of Trade may fix; and (2) the sum of [two guineas], or such less sum as the Board of Trade may fix, as a commuted payment for every day or part of a day that the surveyor is occupied in inspecting or testing the material, or in travelling to and from the place of inspection, to cover loss of time, subsistence, and other expenses.

If no such inspection of materials is made during manufacture, the surveyor is not to give a declaration in the case of steel boilers or machinery unless he has satisfied himself, by requiring a sufficient number of plates or quantity of material to be taken out and tested, or in some other effectual manner, that the material and workmanship are entirely satisfactory, and in no case without a special reference to the Board of Trade.

SUMMARY.

The preceding rules are summarised in the following short table, in which the method has been carried out of indicating all such dimensions as are measured in inches by capitals, such as are measured in sixteenths of an inch by small letters, and all coefficients, &c., by black letters.

C and **C′** = coefficients.
W P = permissible working pressure.
B and **B′** = percentage of joint, respectively of plate and of rivets.
N = number of rivets included within one pitch of external row.
F = factor of safety.
T and t = thickness of plate, measured respectively in inches and in sixteenths.
P and P′ = pitches of rivets, or of stays, or of stay tubes.
D = internal diameter of shells, or of plain tubes; effective diameters of rivets or of stays, or external diameter of furnaces in inches, measured at the plain cylindrical part or at the bottom of the corrugations.
L′ and L″ = length of plain cylindrical parts of furnaces, depth of combination chamber, or length of girders, measured respectively in feet and in inches.
H = depth of girders measured in inches.
A = sectional area of stays or of rivets in square inches.

RIVETED JOINTS. (See appended Tables.)

Percentage of plate $\mathbf{B} = \dfrac{P - D}{P}$.

Percentage of rivets $\mathbf{B}' = \mathbf{C} \cdot \mathbf{N} \cdot \dfrac{\mathbf{A}}{P \cdot T}$, or $\mathbf{C}' \cdot \mathbf{N} \cdot \dfrac{D^2}{P \cdot T}$.

TABLE OF COEFFICIENTS

Material of: Plate	Material of: Rivet	Double Butt Straps C	Double Butt Straps C′	Lap Joints C	Lap Joints C′
Iron, punched .	**Iron**	175·0	137·4	100·0	78·5
„ drilled . .	„ . .	175·0	137·4	100·0	78·5
Steel, „ . .	„ . .	107·6	84·6	61·5	48·3
„ „ . .	**Steel**	143·6	112·8	82·2	64·4

The coefficients for determining the working pressure of the shell depend on the material and on conditions of workmanship, the ultimate strength of the material being divided by the sum of 4·5 and one or both of the following additions :—

Table of Additions to the Factors of Safety.

Type of Joint	Longitudinal Seams				Circumferential Seams				
	Double Butt Straps	Lap Joint			Double Butt Straps		Lap Joint		
Riveting	Any Sort	Single	Double	Treble	Single[1]	Double	Single[1]	Double[1]	Treble
Holes punched before bending .	0·5	1·5	0·7	0·6	0·3	0·2	0·4	0·3	0·2
„ „ after „ .	0·3	1·3	0·5	0·4	0·25	0·15	0·35	0·25	0·15
„ drilled before „ .	0·3	1·3	0·5	0·4	0·25	0·15	0·35	0·25	0·15
„ „ after „ .	0·15	1·15	0·35	0·25	0·2	0·10	0·3	0·2	0·1
„ „ in place „ .	0·0	1·0	0·2	0·1	0·1	0·0	0·2	0·1	0·0

For good but not tested **iron** add ·5.
If there are no circumferential seams add ·4.
[1] In these three cases ·2 has to be added for double-ended boilers, but only for the central seam.

Iron Boiler Shells. $$WP = \frac{C}{F} \cdot \frac{(B \text{ or } B')}{50} \cdot \frac{T}{D}.$$

The value of **C** is

40,000 for wrought **iron** (across the grain), not tested.
47,000 „ „ „ (with the grain) „

If tested, C is the lowest ultimate tensile strength in lbs. per sq. inch. For superheaters use 30,000 and 22,400.

Steel Boiler Shells. $$WP = \frac{C}{F} \cdot \frac{B}{50} \cdot \frac{T}{D}, \text{ or } \frac{C \cdot B' \cdot T}{225 \cdot D}.$$

For **steel** of less than 32 tons the values of **C** are

60,480 for **steel** of more than 27 tons.
62,720 „ „ „ 28 „
64,960 „ „ „ 29 „
67,200 „ „ „ 30 „

FURNACES. (See appended Tables.)

Plain Furnaces. $$\mathbf{W\,P} = \frac{C \,.\, T^2 \,.}{D \,.\, (L' + 1)}, \text{ or } \frac{C' \,.\, t^2}{D \,.\, (L' + 1)}.$$

TABLE OF VALUES OF C and C'.

Material of Furnace	C			C'		
	Iron		Steel	Iron		Steel
Longitudinal Seams of Furnace	Drilled	Punched	Drilled	Drilled	Punched	Drilled
Single butt straps double-riveted, or double butt straps single-riveted . .	90,000	85,000	99,000	351·6	332·0	386·7
Single butt straps single-riveted, or lap-bevelled and double-riveted	80,000	75,000	88,000	312·5	293·0	343·7
Lap not bevelled, double-riveted	75,000	70,000	82,500	293·0	273·4	322·2
Lap bevelled, single-riveted .	70,000	65,000	77,000	273·4	255·9	300·8
Lap not bevelled, single-riveted	65,000	60,000	71,500	253·9	234·4	279·3
Longitudinal seams welded .	90,000		99,000	351·6		386·7

In no case should the working pressure exceed the values found by the following formulæ :—

$$\mathbf{W\,P} = \frac{8{,}000\ T}{D} = \frac{500\ t}{D} \text{ for iron.}$$

$$\mathbf{W\,P} = \frac{8{,}800\ T}{D} = \frac{550\ t}{D} \text{ for steel.}$$

For short lengths of flanged furnaces use the following formulæ :—

$$\left.\begin{aligned}\mathbf{W\,P} &= \frac{8{,}000}{D}\left(\frac{5}{3} \,.\, T - \frac{L''}{202{\cdot}5} - \frac{1}{16{,}875}\right) \\ &= \frac{500}{D} \,.\, \left(\frac{5}{3} \,.\, t - L'' \,.\, {\cdot}07902 - {\cdot}9481\right)\end{aligned}\right\} \text{ for iron plates.}$$

$$\left.\begin{aligned}\mathbf{W\,P} &= \frac{8{,}800}{D} \,.\, \left(\frac{5}{3} \,.\, T - \frac{L''}{202{\cdot}5} - \frac{1}{16{\cdot}875}\right) \\ &= \frac{550}{D} \,.\, \left(\frac{5}{3} \,.\, t - L'' \,.\, {\cdot}07902 - {\cdot}9481\right)\end{aligned}\right\} \text{ for steel plates.}$$

These formulæ can only be used as long as $L'' \leqq 135\,T - 12$.

The working pressures for various patent furnaces are found by the following formulæ :—

Fox's corrugated iron furnaces, $\mathbf{W\,P} = 9{,}000 \,.\, \frac{T}{D} = 562{\cdot}5 \,.\, \frac{t}{D}$.

Fox's corrugated steel furnaces / Purves's ribbed „ „ } $\mathbf{W\,P} = 14{,}000 \,.\, \frac{T}{D} = 875 \,.\, \frac{t}{D}$.

Flat Plate. (See appended Tables.)

$$\mathbf{WP}. = \frac{\mathbf{C}\,.\,(t+1)^2}{(\mathbf{P}\,.\,\mathbf{P}'-6)}.$$

If the pitches P and P′ are equal, then

$$\mathbf{P} = \sqrt{6 + \frac{\mathbf{C}\,.\,(t+1)^2}{\mathbf{W\,P}}}.$$

For unequal pitches (see T. W. Trail, 1890, pp. 67, 199) apparently

$$\mathbf{W\,P} = \frac{2\,\mathbf{C}\,.\,(t+1)^2}{p^2-12},$$

where p = diagonal pitch.

Table of Values of C.

Material: Particulars about Stays	Material: Position of Plates	Iron Plates: Combustion Chambers	Iron Plates: Inside of Uptake	Iron Plates: External Plates	Steel Plates: Combustion Chambers	Steel Plates: Inside of Uptake	Steel Plates: External Plates
Stay ends riveted		60	36	...	66	39·6	...
„ nutted		80	54	90	100	67·5	112·5
Double nuts and washers 3 D × $\frac{2}{3}$ T		...	60	100	...	75	125·0
„ „ „ $\frac{2}{3}$ P × T		...	...	150	...	...	187·5
„ „ bars $\frac{2}{3}$ P × T		...	...	160	...	...	200·0

Tube Plates.

$$\mathbf{W\,P} = \frac{\mathbf{C}\,.\,(\mathrm{P}-\mathrm{D})\,\mathrm{T}}{\mathrm{L}''.\,\mathrm{P}} = \frac{\mathbf{C}'\,.\,(\mathrm{P}-\mathrm{D})\,t}{\mathrm{L}''\,.\,\mathrm{P}}.$$

Table of Values of C and C′.

Material	Iron	Steel
C	16,000	20,000
C′	1,000	1,250

STAYS. (See appended Tables.)

$$\mathbf{WP} = \mathbf{C}\,\frac{\mathbf{A}}{\mathbf{P}\,.\,\mathbf{P}'}, \text{ or } \mathbf{C}'\,\frac{D^2}{\mathbf{P}\,.\,\mathbf{P}'}.$$

TABLE OF VALUES OF C AND C'.

Material of Stays	C	C'
Welded or heated iron stays	5,000	3,925
Iron stays	7,000	5,498
Steel „	9,000	7,069

GIRDERS. $$\mathbf{WP} = \frac{\mathbf{C}\,.\,H^2\,T}{L''\,.\,(L''-P)\,.\,P'}.$$

TABLE OF VALUES OF C.

Number of Stays in Girders	Material of Girders	
	Iron	Steel
One stay	6,000	6,600
Two or three stays	9,000	9,900
Four or five „	10,200	11,220

TABLES.

When used in the drawing office, it is strongly recommended that those parts of the following tables which are inapplicable to the particular works should be obliterated.

TABLE FOR FINDING THE DIAMETERS OF RIVETS AND OF PITCH IN RIVETED JOINTS.

A short table will be found on p. 261 which shows the smallest permissible percentage for any particular joint; the conditions are fully explained and illustrated.

The following table contains the values of $\mathbf{N}\,.\,D \div T$ = number of rivets $\times$ diameter of rivets $\div$ thickness of steel plate. Having found the number in the table corresponding to the desired percentage, it has to be divided by the number of rivets and multiplied by the thickness of the plate. The result is the diameter of the rivet hole. This has to be multiplied by the corresponding value in the first column of this table, viz. pitch $\div$ rivet diameter, and the product is the pitch of the rivet. For examples see p. 283.

This is only true if the factor of safety is 4·5. For any other factor $\mathbf{F}$ the value $\mathbf{N}\,.\,D \div T$ has to be multiplied by $\frac{4{\cdot}5}{\mathbf{F}}$.

The pitch is limited by the formula $P = 1\frac{5}{8} + \mathbf{C} \times T$, and may not exceed 10″. (See p. 302.)

MINIMUM VALUES OF $N \cdot D \div T = \dfrac{\text{Number} \times \text{Diameter of Rivets}}{\text{Thickness of Plate}}$.

Pitch ÷ Rivet Diameter	Percentage of Plate	LAP JOINT: Iron Plates Iron Rivets	LAP JOINT: Steel Plates Steel Rivets	LAP JOINT: Steel Plates Iron Rivets	BUTT STRAPS: Iron Plates Iron Rivets	BUTT STRAPS: Steel Plates Steel Rivets	BUTT STRAPS: Steel Plates Iron Rivets
	Constants	100	$\frac{23}{28}\cdot100$	$\frac{8}{13}\cdot100$	$\frac{7}{4}\cdot100$	$\frac{7}{4}\cdot\frac{23}{28}\cdot100$	$\frac{7}{4}\cdot\frac{8}{13}\cdot100$
2·50	60	...	2·32	3·10	...	...	...
2·56	61	...	2·42	3·24	...	...	...
2·63	62	2·08	2·53	3·38	...	...	...
2·70	63	2·17	2·64	3·52	...	...	2·01
2·78	64	2·26	2·76	3·68	...	...	2·10
2·86	65	2·36	2·88	3·84	...	...	2·19
2·94	66	2·47	3·01	4·02	...	...	2·29
3·03	67	2·59	3·15	4·20	...	...	2·40
3·12	68	2·71	3·29	4·40	...	...	2·51
3·22	69	2·83	3·45	4·61	...	...	2·63
3·33	70	2·97	3·62	4·83	...	2·07	2·76
3·45	71	3·12	3·79	5·07	...	2·17	2·89
3·57	72	3·27	3·99	5·32	...	2·28	3·04
3·70	73	3·44	4·19	5·59	...	2·39	3·20
3·85	74	3·62	4·41	5·89	2·07	2·52	3·37
4·00	75	3·82	4·65	6·21	2·18	2·66	3·55
4·08	75½	3·92	4·78	6·38	2·24	2·73	3·64
4·17	76	4·03	4·91	6·55	2·30	2·80	3·74
4·26	76½	4·14	5·05	6·74	2·37	2·88	3·85
4·35	77	4·26	5·19	6·93	2·44	2·97	3·96
4·45	77½	4·38	5·34	7·13	2·51	3·05	4·07
4·55	78	4·51	5·49	7·34	2·58	3·14	4·19
4·65	78½	4·65	5·66	7·55	2·66	3·23	4·32
4·76	79	4·79	5·83	7·78	2·74	3·33	4·45
4·88	79½	4·94	6·01	8·02	2·82	3·43	4·58
5·00	80	5·09	6·20	8·28	2·91	3·54	4·73
5·13	80½	5·26	6·40	8·54	3·00	3·66	4·88
5·26	81	5·43	6·61	8·82	3·10	3·78	5·04
5·40	81½	5·61	6·83	9·11	3·21	3·90	5·21
5·55	82	5·80	7·06	9·43	3·32	4·03	5·39
5·71	82½	6·00	7·31	9·75	3·43	4·17	5·57
5·88	83	6·22	7·57	10·10	3·55	4·32	5·77
6·06	83½	6·44	7·84	...	3·68	4·48	5·98
6·25	84	6·68	8·14	...	3·82	4·65	6·21
6·45	84½	6·94	8·45	...	3·97	4·83	6·45
6·67	85	7·22	8·78	...	4·12	5·02	6·70
6·82	85⅓	7·41	9·02	...	4·23	5·15	6·88
6·98	85⅔	7·61	9·26	...	4·35	5·29	7·07
7·14	86	7·82	9·52	...	4·47	5·44	7·26
7·32	86⅓	8·04	9·79	...	4·60	5·60	7·47
7·50	86⅔	8·28	10·07	...	4·73	5·76	7·68
7·69	87	8·52	...	...	4·87	5·93	7·91
7·89	87⅓	8·78	...	...	5·02	6·11	8·15
8·10	87⅔	9·05	...	...	5·17	6·30	8·40
8·33	88	9·34	...	...	5·34	6·50	8·67
8·57	88⅓	9·64	...	...	5·51	6·71	8·95
8·82	88⅔	9·96	...	...	5·69	6·93	9·25
9·09	89	10·30	...	...	5·89	7·17	9·57
9·37	89⅓	...	...	...	6·09	7·42	9·90
9·68	89⅔	...	...	...	6·31	7·69	10·26
10·00	90	...	...	...	6·55	7·97	...

The following tables contain the **internal** diameters of furnaces, calculated according to the Board of Trade Rules :—

INTERNAL DIAMETERS OF CIRCULAR FURNACES IN INCHES.

TYPE	PLAIN FLUES (Iron)						PLAIN FLUES (Steel)						ADAMSON'S RINGS			FOX & PURV.
Formula	$90{,}000 \cdot \frac{T^2}{D\cdot(L'+1)}$					$8{,}000 \cdot \frac{T}{D}$	$99{,}000 \cdot \frac{T^2}{D\cdot(L'+1)}$					$8{,}800 \cdot \frac{T}{D}$	$\frac{550}{D}\left(\text{[illegible]}\,t - L''\cdot 079 - \cdot 948\right)$			$C \cdot \frac{T}{D}$
L. or C.	9 Ft.	8 Ft.	7 Ft.	6 Ft.	5 Ft.		9 Ft.	8 Ft.	7 Ft.	6 Ft.	5 Ft.		36 In.	24 In.	18 In.	14,000
Plate Thickness. Inches.	**60 lbs. Working Pressure.**															
3/8	...	...	25·6	29·4	34·4	49·2	...	...	28·2	32·4	37·9	54·2	56·0	64·8	69·2	86·7
13/32	...	26·7	30·1	34·5	40·4	53·3	26·4	29·4	33·2	38·1	44·6	...	...	...	...	...
7/16	27·8	31·0	35·0	40·1	47·0	...	30·7	34·2	38·6	44·2	51·7	...	...	...	...	...
15/32	32·0	35·7	40·2	46·1	54·0	...	35·3	39·3	44·4	50·8	...	...	...	...	...	...
1/2	36·5	40·6	45·9	52·5	...	...	40·2	44·8	50·6	...	...	...	...	...	...	...
17/32	41·3	46·0	51·8	...	...	...	45·5	50·7	...	...	...	...	...	...	...	...
9/16	46·3	51·6	...	...	...	...	51·1	...	...	...	...[1]	...	...	...	...	...
19/32	51·7	...	...	...	...	...	...	...	...	...	...[1]	...	...	...	...	...
5/8	...	...	...	...	...	...	...	...	...	...[1]	...[1]	...	...	...	...	...
	80 lbs. Working Pressure.															
3/8	...	...	...	...	25·6	36·7	...	...	...	...	28·2	40·5	41·8	48·4	51·6	64·9
13/32	...	...	...	25·7	30·1	39·8	...	...	...	28·4	33·2	43·9	47·6	54·1	...	...
7/16	...	...	26·0	29·9	35·0	46·0	...	...	28·7	32·9	38·6	47·2	53·3	...	...	...
15/32	...	26·4	30·0	34·4	40·3	49·1	26·1	29·1	33·0	37·9	44·4	50·6	...	...	...	...
1/2	27·1	30·2	34·2	39·2	45·9	52·1	29·9	33·4	37·7	43·2	50·6	...	...	...	...	...
17/32	30·7	34·2	38·6	44·3	51·9	...	33·8	37·7	42·6	48·8	...	...	...	...	...	...
9/16	34·5	38·4	43·4	49·7	...	...	38·0	42·4	47·8	54·8	...	...	...	...	...	...
19/32	38·5	42·9	48·4	55·5	...	...	42·4	47·3	53·3	...	...	...	...	...	...	...
5/8	42·7	47·5	53·7	...	...	...	47·1	50·9	...	...	...	...	...	...	...	...
	100 lbs. Working Pressure.															
3/8	...	...	...	...	...	29·2	...	...	...	...	...	32·2	33·3	38·5	41·2	51·7
13/32	...	...	...	...	...	31·7	...	...	...	...	26·3	34·9	37·9	43·1	45·7	...
7/16	...	...	...	...	27·8	34·1	...	...	...	26·2	30·7	37·6	42·4	47·6	50·3	...
15/32	...	...	...	27·3	32·0	36·6	...	...	26·2	30·1	35·5	40·3	46·9	52·1	...	...
1/2	...	24·0	27·1	31·1	36·5	39·0	...	26·5	29·9	34·3	40·2	43·0	51·5	...	...	...
17/32	24·2	27·2	30·7	35·2	41·3	41·4	26·8	30·0	33·8	38·8	45·5	45·7	...	...	...	...
9/16	27·4	30·5	34·5	39·6	43·8	43·8	30·2	33·7	38·0	43·6	50·1	48·4	...	...	...	...
19/32	30·5	34·1	38·4	44·1	46·3	46·3	33·7	37·6	42·4	48·7	54·7	51·1	...	...	...	...
5/8	33·9	36·7	42·7	48·7	48·7	48·7	38·4	41·5	47·1	54·1	59·3	53·8	...	...	...	...
	120 lbs. Working Pressure.															
3/8	...	...	...	...	...	24·2	...	...	...	...	...	26·7	27·6	31·9	34·2	43·0
13/32	...	...	...	...	...	26·3	...	...	...	...	...	29·0	31·4	35·7	38·0	46·6
7/16	...	...	...	...	...	28·3	...	...	...	...	...	31·2	35·2	39·5	41·7	50·2
15/32	...	...	...	...	26·5	30·3	...	...	...	...	29·2	33·4	39·0	43·3	45·5	...
1/2	...	...	...	25·8	30·2	32·3	...	...	...	28·5	33·4	35·7	42·7	47·1	49·2	...
17/32	...	...	25·4	29·2	34·2	34·4	...	...	28·0	32·2	37·7	37·9	46·5	50·8	53·0	...
9/16	...	25·2	28·5	32·8	36·4	36·4	...	27·9	31·5	36·1	41·6	40·1	50·2	...	...	...
19/32	25·2	28·2	31·9	36·6	38·4	38·4	27·9	31·1	35·2	42·0	45·4	42·4	...	...	...	...
5/8	28·0	31·4	35·4	40·4	40·4	40·4	31·0	34·5	39·0	44·8	49·2	44·6	...	...	...	...

[1] In every one of the following furnace tables the diameters for these four dimensions are estimated with the help of the formula over the last column but one. See also foot note, p. 287.

INTERNAL DIAMETERS OF CIRCULAR FURNACES IN INCHES—*cont.*

TYPE	PLAIN FLUES (Iron)						PLAIN FLUES (Steel)						ADAMSON'S RINGS			FOX & PURV.
Formula	$90{,}000 \cdot \frac{T^2}{D.(L'+1)}$					$\frac{T}{D}$	$99{,}000 \cdot \frac{T^2}{D.(L'+1)}$					$\frac{T}{D}$	$\frac{550}{D}$ ([illegible]$t - L'' \cdot 079 - \cdot 948$)			$C.\frac{T}{D}$
L. or C.	9 Ft.	8 Ft.	7 Ft.	6 Ft.	5 Ft.	8,000	9 Ft.	8 Ft.	7 Ft.	6 Ft.	5 Ft.	8,800	36 In.	24 In.	18 In.	14,000
Plate Thickness. Inches	**140 lbs. Working Pressure.**															
$\frac{3}{8}$	...	...	...	...	...	...	...	...	...	...	...	...	...	27·3	29·2	36·7
$\frac{13}{32}$	...	...	...	...	...	...	...	...	...	...	...	24·7	26·8	30·6	32·4	39·8
$\frac{7}{16}$	...	...	...	...	...	24·1	..	...	...	...	...	26·6	30·1	33·8	35·6	42·9
$\frac{15}{32}$	...	...	...	...	...	25·9	...	...	...	...	...	28·5	33·3	37·0	38·8	45·9
$\frac{1}{2}$	...	...	...	...	25·8	27·6	...	...	...	...	28·5	30·4	36·5	40·2	42·1	49·0
$\frac{17}{32}$	...	...	...	24·9	29·2	29·3	...	...	...	27·4	32·2	32·3	39·7	43·4	45·3	52·1
$\frac{9}{16}$	...	...	24·3	27·9	31·0	31·0	...	...	26·8	30·8	35·5	34·2	45·9	46·6	48·5	...
$\frac{19}{32}$	...	...	27·1	31·2	32·7	32·7	...	26·4	30·0	34·4	38·8	36·1	46·1	49·8	51·7	...
$\frac{5}{8}$	...	25·8	30·1	34·5	34·5	34·5	26·4	28·5	33·3	38·3	41·0	38·0	49·3	53·1	...	...
	160 lbs. Working Pressure.															
$\frac{3}{8}$	...	...	...	...	...	...	...	...	...	...	...	...	...	...	25·5	32·1
$\frac{13}{32}$	...	...	...	...	...	...	...	...	...	...	...	...	...	26·6	28·3	34·7
$\frac{7}{16}$	...	...	...	...	...	...	...	...	...	...	...	...	26·2	29·5	31·1	37·4
$\frac{15}{32}$	...	...	...	...	...	...	...	...	...	...	...	24·8	29·1	32·3	33·9	40·1
$\frac{1}{2}$	...	...	...	...	...	24·0	...	...	...	...	...	26·5	31·8	35·3	36·7	42·7
$\frac{17}{32}$	...	...	...	...	25·5	25·5	...	...	...	...	28·0	28·2	34·6	37·9	39·5	45·4
$\frac{9}{16}$	...	...	...	24·3	27·0	27·0	...	...	...	26·8	30·9	29·8	37·4	40·7	42·3	48·1
$\frac{19}{32}$	...	...	...	27·1	28·5	28·5	...	...	25·1	30·0	33·8	31·5	40·2	43·5	45·1	50·8
$\frac{5}{8}$	...	...	26·2	30·0	30·0	30·0	24·3	25·5	28·9	33·4	36·6	33·1	43·0	46·3	47·9	...
	180 lbs. Working Pressure.															
$\frac{3}{8}$	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	28·4
$\frac{13}{32}$	...	...	...	...	...	...	...	...	...	...	...	...	...	...	25·0	30·8
$\frac{7}{16}$	...	...	...	...	...	...	...	...	...	...	...	...	...	26·1	27 5	33·1
$\frac{15}{32}$	...	...	...	...	...	...	...	...	...	...	...	...	25·7	28·6	30·0	35·5
$\frac{1}{2}$	...	...	...	...	...	...	...	...	...	...	...	...	28·1	31·1	32·5	37·9
$\frac{17}{32}$	...	...	...	...	...	...	...	...	...	...	...	24·9	30·6	33·5	35·0	40·2
$\frac{9}{16}$	...	...	...	...	...	...	...	...	...	...	27·3	26·4	33·1	36·0	37·4	42·6
$\frac{19}{32}$	...	...	...	25·0	25·2	25·2	...	...	...	27·5	28·9	28·9	35·6	38·5	39·9	45·0
$\frac{5}{8}$	...	...	...	26·5	26·5	26·5	...	...	25·6	29·5	32·4	29·3	38·1	41·0	42·4	47·4
	200 lbs. Working Pressure.															
$\frac{3}{8}$	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	25·5
$\frac{13}{32}$	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	27·6
$\frac{7}{16}$	...	...	...	...	...	...	...	...	..	...	...	...	...	...	...	29·7
$\frac{15}{32}$	...	...	...	...	...	...	...	...	...	...	...	...	...	25·7	26·9	31·9
$\frac{1}{2}$	...	...	...	...	...	...	...	...	...	...	...	...	25·4	27·9	29·1	34·0
$\frac{17}{32}$	...	...	...	...	...	...	...	...	...	...	...	...	27·5	30·1	31·4	36·1
$\frac{9}{16}$	...	...	...	...	...	...	...	...	...	...	24·5	...	29·7	32·3	33·6	38·2
$\frac{19}{32}$	...	...	...	...	...	...	...	...	...	...	26·8	24·9	31·9	34·5	35·8	40·4
$\frac{5}{8}$	...	...	...	...	23·7	23·7	...	...	...	26·5	29·1	26·2	34·1	36·8	38·1	42·5

Tables for Finding Pitches of Stays.

The following tables contain the maximum permissible pitches of stays in flat plates. These tables are only correct for such cases in which the two pitches are equal. As long as they are nearly equal—say, within 20 % of each other—it is sufficiently correct to take the geometrical mean pitch. When the difference is greater certain restrictions come into force (see p. 324). When the pitches are less than six inches there are further restrictions. (See T. W. Trail, 1890, p. 29.)

Maximum Square Pitch in Inches of Stayed Flat Plates.

	Stayed Iron Plates							Stayed Steel Plates						
	Heating Surfaces		External Plates					Heating Surfaces		External Plates				
Constants	60	80	67·5	90	100	150	160	66	100	75	112·5	125	187·5	200
Plate Thickness	Stay Ends Riveted	Stay Ends Nutted	Stay Ends Riveted	Stay Ends Nutted	Nuts & Washers $3d \times \frac{2}{3}t$	Do. Riveted $\frac{2}{3}P \times t$	Strips Riveted $\frac{2}{3}P \times t$	Stay Ends Riveted	Stay Ends Nutted	Stay Ends Riveted	Stay Ends Nutted	Nuts & Washers $3d \times \frac{2}{3}t$	Do. Riveted $\frac{2}{3}P \times t$	Strips Riveted $\frac{2}{3}P \times t$
Inches	**60 lbs. Working Pressure.**													
$\frac{3}{8}$	7·41	8·44	7·81	8·91	9·36	11·3	11·7	7·74	9·36	8·20	9·89	10·4	12·4	13·0
$\frac{7}{16}$	8·36	9·55	8·83	10·1	10·6	12·9	13·3	8·74	10·6	9·27	11·2	11·8	14·3	14·8
$\frac{1}{2}$	9·32	10·7	9·85	11·3	11·9	14·4	14·9	9·75	11·9	10·3	12·6	13·2	16·0	16·6
$\frac{9}{16}$	10·3	11·8	10·9	12·5	13·1	16·0	16·5	10·8	13·1	11·4	13·9	14·6	17·8	18·4
$\frac{5}{8}$	11·3	12·9	11·9	13·7	14·4	17·6	18·1	11·8	14·4	12·5	15·3	16·1	19·6	20·2
$\frac{11}{16}$	12·2	14·1	13·0	14·9	15·7	19·1	19·7	12·8	15·7	13·6	16·6	17·5	21·4	...
$\frac{3}{4}$	13·2	15·2	14·0	16·1	17·0	20·7	21·3	13·8	17·0	14·7	18·0	18·9	...	...
$\frac{13}{16}$	...	...	15·0	17·3	18·2	...	...	...	...	15·8	19·3	20·3	...	...
$\frac{7}{8}$	...	...	16·1	18·5	19·5	...	...	...	...	16·9	20·7	...	...	...
$\frac{15}{16}$	...	...	17·1	19·7	20·8	...	...	...	...	18·0	...	...	...	...
1	...	...	18·2	21 0	...	...	...	...	...	19·1	...	...	...	...
$1\frac{1}{16}$	...	...	19·2	...	...	...	...	...	...	20·2	...	...	...	...
$1\frac{1}{8}$	...	...	20·3	...	...	...	...	...	...	...	...	...	...	...
	80 lbs. Working Pressure.													
$\frac{3}{8}$	6·53	7·41	6·88	7·81	8·20	9·89	10·2	6·81	8·20	7·20	8·65	9·08	11·0	11·3
$\frac{7}{16}$	7·34	8·36	7·74	8·83	9·27	11·2	11·6	7·66	9·27	8·12	9·79	10·3	12·5	12·9
$\frac{1}{2}$	8·17	9·32	8·62	9·85	10·3	12·6	13·0	8·53	10·3	9·05	10·9	11·5	14·0	14·4
$\frac{9}{16}$	9·00	10·3	9·50	10·9	11·4	13·9	14·3	9·40	11·4	9·98	12·1	12·7	15·5	16·0
$\frac{5}{8}$	9·83	11·3	10·4	11·9	12·5	15·3	15·7	10·3	12·5	10·9	13·3	14·0	17·0	17·6
$\frac{11}{16}$	10·7	12·2	11·3	13·0	13·6	16·6	17·1	11·2	13·6	11·9	14·4	15·2	18·5	19·1
$\frac{3}{4}$	11·5	13·2	12·2	14·0	14·7	18·0	18·5	12·1	14·7	12·8	15·6	16·4	20·0	20·7
$\frac{13}{16}$	...	...	13·1	15·0	15·8	19·3	19·9	...	...	13·8	16·8	17·7	...	...
$\frac{7}{8}$	...	...	14·0	16·1	16·9	20·7	21·3	...	...	14·7	17·9	18·9	...	...
$\frac{15}{16}$	...	...	14·9	17·1	18·0	...	...	...	...	15·7	19·1	20·1	...	...
1	...	...	15·8	18·2	19·2	...	...	...	...	16·6	20·3	...	...	...
$1\frac{1}{16}$	...	...	16·7	19·2	20·3	...	...	...	...	17·6	...	...	...	...
$1\frac{1}{8}$	...	...	17·6	20·3	...	...	...	...	...	18·5	...	...	...	...

	STAYED IRON PLATES							STAYED STEEL PLATES						
	Heating Surfaces		External Plates					Heating Surfaces		External Plates				
Constants	60	80	67·5	90	100	150	160	66	100	75	112·5	125	187·5	200
Plate Thickness	Stay Ends Riveted	Stay Ends Nutted	Stay Ends Riveted	Stay Ends Nutted	Nuts & Washers 3d × ⅔t	Do. Riveted ⅔P × t	Strips Riveted ⅔P × t	Stay Ends Riveted	Stay Ends Nutted	Stay Ends Riveted	Stay Ends Nutted	Nuts & Washers 3d × ⅔t	Do. Riveted ⅔P × t	Strips Riveted ⅔P × t
Inches.	100 LBS. WORKING PRESSURE.													
3/8	5·88	6·72	6·25	7·07	7·41	8·92	9·18	6·19	7·41	6·53	7·81	8·20	9·89	10·2
7/16	6·66	7·56	7·01	7·97	8·36	10·1	10·4	6·94	8·36	7·34	8·83	9·27	11·2	11·6
1/2	7·39	8·41	7·79	8·88	9·32	11·3	11·6	7·71	9·32	8·17	9·85	10·3	12·6	13·0
9/16	8·12	9·27	8·57	9·79	10·3	12·5	12·9	8·48	10·3	9·00	10·9	11·4	13·9	14·3
5/8	8·86	10·1	9·36	10·7	11·3	13·7	14·1	9·26	11·3	9·83	11·9	12·5	15·3	15·7
11/16	9·61	11·0	10·1	11·6	12·2	14·9	15·4	10·0	12·2	10·7	13·0	13·6	16·6	17·1
3/4	10·4	11·9	10·9	12·6	13·2	16·1	16·6	10·8	13·2	11·5	14·0	14·7	18·0	18·5
13/16	...	...	11·8	13·5	14·2	17·3	17·9	...	...	12·4	15·0	15·8	19·3	19·9
7/8	...	...	12·6	14·4	15·2	18·5	19·1	...	...	13·2	16·1	16·9	20·7	21·3
15/16	...	...	13·4	15·4	16·2	19·7	21·4	...	...	14·1	17·1	18·0	...	...
1	...	...	14·2	16·3	17·2	20·9	...	...	...	14·9	18·2	19·2	...	...
1 1/16	...	...	15·0	17·2	18·2	...	...	...	...	15·8	19·2	20·3	...	...
1 1/8	...	...	15·8	18·2	19·1	...	...	...	...	16·6	20·3	...	...	...
	120 LBS. WORKING PRESSURE.													
3/8	5·52	6·21	5·79	6·53	6·84	...	...	5·73	6·84	6·05	7·20	7·55	9·08	9·36
7/16	6·16	6·97	6·48	7·34	7·70	9·27	9·55	6·41	7·70	6·78	8·12	8·52	10·3	10·6
1/2	6·82	7·74	7·18	8·17	8·57	10·3	10·7	7·11	8·57	7·52	9·05	9·50	11·5	11·9
9/16	7·48	8·52	7·89	9·00	9·45	11·4	11·8	7·81	9·45	8·27	9·98	10·5	12·7	13·1
5/8	8·15	9·31	8·60	9·83	10·3	12·5	12·9	8·51	10·3	9·03	10·9	11·5	14·0	14·4
11/16	8·83	10·1	9·32	10·7	11·2	13·6	14·1	9·23	11·2	9·79	11·9	12·5	15·2	15·7
3/4	9·51	10·9	10·0	11·5	12·1	14·7	15·2	9·94	12·1	10·6	12·8	13·5	16·4	17·0
13/16	...	...	10·8	12·4	13·0	15·8	16·3	...	...	11·3	13·8	14·5	17·7	18·2
7/8	...	...	11·5	13·2	13·9	16·9	17·5	...	...	12·1	14·7	15·5	18·9	19·5
15/16	...	...	12·2	14·1	14·8	18·0	18·6	...	...	12·9	15·7	16·5	20·1	20·8
1	...	...	13·0	14·9	15·7	19·2	19·8	...	...	13·7	16·6	17·5	...	...
1 1/16	...	...	13·7	15·8	16·6	20·3	20·9	...	...	14·4	17·6	18·5	...	...
1 1/8	...	...	14·4	16·6	17·5	...	...	...	...	15·2	18·5	19·5	...	...
	140 LBS. WORKING PRESSURE.													
3/8	5·20	5·83	5·44	6·12	6·40	...	...	5·39	6·40	5·68	6·73	7·05	...	...
7/16	5·78	6·52	6·07	6·86	7·19	...	...	6·01	7·19	6·34	7·57	7·94	9·57	9·87
1/2	6·38	7·23	6·71	7·62	7·99	9·63	9·92	6·64	7·99	7·02	8·43	8·85	10·7	11·0
9/16	6·99	7·94	7·36	8·38	8·80	10·6	11·0	7·31	8·80	7·71	9·29	9·76	11·8	12·2
5/8	7·60	8·66	8·02	9·15	9·61	11·6	12·0	7·94	9·61	8·41	10·2	10·7	13·0	13·4
11/16	8·22	9·39	8·68	9·92	10·4	12·7	13·1	8·59	10·4	9·11	11·0	11·6	14·1	14·5
3/4	8·85	10·1	9·35	10·7	11·2	13·7	14·1	9·25	11·2	9·82	11·9	12·5	15·2	15·7
13/16	...	...	10·0	11·5	12·1	14·7	15·2	...	...	10·5	12·8	13·4	16·4	16·9
7/8	...	...	10·7	12·3	12·9	15·7	16·2	...	...	11·2	13·7	14·4	17·5	18·1
15/16	...	...	11·4	13·1	13·7	16·7	17·3	...	...	12·0	14·5	15·3	18·7	19·3
1	...	...	12·0	13·8	14·6	17·8	18·3	...	...	12·7	15·4	16·2	19·8	20·5
1 1/16	...	...	12·7	14·6	15·4	18·8	19·4	...	...	13·4	16·3	17·2	21·0	...
1 1/8	...	...	13·4	15·4	16·2	19·8	20·5	...	...	14·1	17·2	18·1	...	...

	STAYED IRON PLATES							STAYED STEEL PLATES						
	Heating Surfaces		External Plates					Heating Surfaces		External Plates				
Constants	60	80	67·5	90	100	150	160	66	100	75	112·5	125	187·5	200
Plate Thickness	Stay Ends Riveted	Stay Ends Nutted	Stay Ends Riveted	Stay Ends Nutted	Nuts & Washers 3d×⅔t	Do. Riveted ⅔P×t	Strips Riveted ⅔P×t	Stay Ends Riveted	Stay Ends Nutted	Stay Ends Riveted	Stay Ends Nutted	Nuts & Washers 3d×⅔t	Do. Riveted ⅔P×t	Strips Riveted ⅔P×t
Inches.							160 LBS. WORKING PRESSURE.							
3/8	4·94	5·52	5·16	5·79	6·05	...	...	5·12	6·05	5·38	6·36	6·65	...	...
7/16	5·48	6·16	5·74	6·48	6·78	...	...	5·69	6·78	6·00	7·14	7·48	9·00	9·27
1/2	6·03	6·82	6·33	7·18	7·52	9·05	9·32	6·27	7·52	6·63	7·93	8·32	10·0	10·3
9/16	6·59	7·48	6·94	7·89	8·27	9·98	10·3	6·87	8·27	7·27	8·73	9·17	11·1	11·4
5/8	7·16	8·15	7·55	8·60	9·03	10·9	11·3	7·47	9·03	7·92	9·54	10·0	12·1	12·5
11/16	7·74	8·83	8·17	9·32	9·79	11·9	12·2	8·08	9·79	8·57	10·3	10·9	13·2	13·6
3/4	8·33	9·51	8·79	10·0	10.6	12·8	13·2	8·70	10·6	9·23	11·2	11·7	14·3	14·7
13/16	...	...	9·41	10·8	11·3	13·8	14·2	...	...	9·89	12·0	12·6	15·3	15·8
7/8	...	...	10·0	11·5	12·1	14·7	15·2	...	...	10·5	12·8	13·5	16·4	16·9
15/16	...	...	10·7	12·2	12·9	15·7	16·2	...	...	11·2	13·6	14·3	17·5	18·0
1	...	...	11·3	13·0	13·7	16·6	17·2	...	...	11·9	14·5	15·2	18·6	19·2
1 1/16	...	...	11·9	13·7	14·4	17·6	18·2	...	...	12·6	15·3	16·1	19·6	20·3
1 1/8	...	...	12·6	14·4	15·2	18·6	19·1	...	...	13·2	16·1	17·0	20·7	...
							180 LBS. WORKING PRESSURE.							
3/8	4·73	5·27	4·44	5·52	5·77	...	...	4·90	5·77	5·14	6·05	6·32	...	...
7/16	5·23	5·87	5·48	6·16	6·44	...	...	5·43	6·44	5·72	6·78	7·10	...	...
1/2	5·57	6·48	5·99	6·82	7·14	...	...	5·97	7·14	6·30	7·52	7·89	9·50	9·79
9/16	6·27	7·10	6·59	7·48	7·84	9·45	9·74	6·53	7·84	6·90	8·27	8·68	10·5	10·8
5/8	6·80	7·73	7·16	8·15	8·55	10·3	10·6	7·09	8·55	7·51	9·03	9·48	11·5	11·8
11/16	7·34	8·37	7·74	8·83	9·27	11·2	11·6	7·66	9·27	8·12	9·79	10·3	12·5	12·9
3/4	7·89	9·00	8·33	9·51	9·99	12·1	12·5	8·24	9·99	8·74	10·6	11·1	13·5	13·9
13/16	...	...	8·91	10·2	10·7	13·0	13·4	...	...	9·36	11·3	11·9	14·5	15·0
7/8	...	...	9·50	10·9	11·4	13·9	14·3	...	...	9·98	12·1	12·7	15·5	16·0
15/16	...	...	10·1	11·6	12·2	14·8	15·3	...	...	10·6	12·9	13·5	16·5	17·0
1	...	...	10·7	12·3	12·9	15·7	16·2	...	...	11·2	13·7	14·4	17·5	18·1
1 1/16	...	...	11·3	13·0	13·6	16·6	17·1	...	...	11·9	14·4	15·2	18·5	19·1
1 1/8	...	...	11·9	13·6	14·4	17·5	18·1	...	...	12·5	15·2	16·0	19·5	20·2
							200 LBS. WORKING PRESSURE.							
3/8	4·55	5·06	4·75	5·30	5·52	...	...	4·71	5·52	4·94	5·79	6·05	...	...
7/16	5·02	5·60	5·26	5·90	6·16	...	...	5·21	6·16	5·48	6·48	6·78	...	...
1/2	5·50	6·19	5·77	6·51	6·82	...	...	5·72	6·82	6·03	7·18	7·52	9·05	9·32
9/16	6·00	6·78	6·30	7·14	7·48	9·00	9·27	6·24	7·48	6·59	7·89	8·27	9·98	10·3
5/8	6·50	7·37	6·84	7·77	8·15	9·83	10·1	6·77	8·15	7·16	8·60	9·03	10·9	11·3
11/16	7·01	7·97	7·39	8·41	8·83	10·7	11·0	7·31	8·83	7·74	9·32	9·79	11·9	12·2
3/4	7·53	8·58	7·94	9·05	9·51	11·5	11·8	7·86	9·51	8·33	10·0	10·6	12·8	13·2
13/16	...	...	8·49	9·70	10·2	12·4	12·7	...	...	8·91	10·8	11·3	13·8	14·2
7/8	...	...	9·05	10·3	10·9	13·2	13·6	...	...	9·50	11·5	12·1	14·7	15·2
15/16	...	...	9·61	11·0	11·6	14·1	14·5	...	...	10·1	12·2	12·9	15·7	16·2
1	...	...	10·2	11·7	12·3	14·9	15·4	...	...	10·7	13·0	13·7	16·6	17·2
1 1/16	...	...	10·7	12·3	13·0	15·8	16·3	...	...	11·3	13·7	14·4	17·6	18·2
1 1/8	...	...	11·3	13·0	13·6	16·6	17·2	...	...	11·9	14·4	15·2	18·6	19·1

The following two tables contain the maximum permissible load to be carried by screwed iron or steel stays :—

PERMISSIBLE WORKING LOADS OF IRON STAYS WITH A STRESS OF 7,000 LBS. PER SQUARE INCH.

Outside Diameters	Working Loads for Plus Threads	Whitworth Threads: Number of Threads	Whitworth Threads	Number of Threads per Inch: 6	7	8	9	10	11	12
			Working Loads of Screwed Stays at 7,000 lbs. per Square Inch							
Inches	Lbs.		Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
$\frac{3}{4}$	3,093	10	2,127	...	...	...	...	2,127	2,208	2,275
$\frac{7}{8}$	4,209	9	2,952	...	...	...	2,952	3,068	3,165	3,245
1	5,498	8	3,879	...	...	3,879	4,045	4,180	4,293	4,387
$1\frac{1}{8}$	6,958	7	4,880	...	4,880	5,120	5,310	5,465	5,594	5,701
$1\frac{1}{4}$	8,590	7	6,262	...	6,262	6,532	6,747	6,921	7,065	7,186
$1\frac{3}{8}$	10,394	6	7,419	7,419	7,813	8,116	8,355	8,549	8,710	8,844
$1\frac{1}{2}$	12,370	6	9,102	9,102	9,539	9,871	10,136	10,349	10,526	10,673
$1\frac{5}{8}$	14,517	5	10,303	10,956	11,435	11,800	12,086	12,321	12,514	12,674
$1\frac{3}{4}$	16,837	5	12,271	12,983	13,503	13,899	14,212	14,464	14,673	14,847
$1\frac{7}{8}$	19,330	$4\frac{1}{2}$	13,910	15,181	15,743	16,170	16,508	16,779	17,004	17,191
2	21,991	$4\frac{1}{2}$	16,182	17,550	18,154	18,614	18,976	19,266	19,508	19,707
$2\frac{1}{8}$	24,826	$4\frac{1}{2}$	18,623	20,092	20,738	21,228	21,616	21,925	22,182	22,396
$2\frac{1}{4}$	27,833	4	20,479	22,806	23,493	24,016	24,426	24,756	25,028	25,255
$2\frac{3}{8}$	31,011	4	23,217	25,691	26,421	26,973	27,408	27,758	28,048	28,287
$2\frac{1}{2}$	34,362	4	26,127	28,748	29,520	30,103	30,564	30,932	31,238	31,491
$2\frac{5}{8}$	37,884	4	29,210	31,976	32,791	33,406	33,888	34,279	34,601	34,866
$2\frac{3}{4}$	41,578	4	32,464	35,377	36,233	36,880	37,361	37,796	38,135	38,413
$2\frac{7}{8}$	45,443	$3\frac{1}{2}$	34,616	38,950	39,848	40,526	41,059	41,487	41,840	42,131
3	49,480	$3\frac{1}{2}$	38,149	42,694	43,634	44,344	44,900	45,347	45,718	46,022
$3\frac{1}{8}$	53,690	...	...	46,611	47,592	48,334	48,914	49,380	49,768	50,085
$3\frac{1}{4}$	58,071	$3\frac{1}{4}$	44,844	50,697	51,719	52,494	53,100	53,586	53,990	54,321
$3\frac{3}{8}$	62,624	...	...	54,958	56,023	56,826	57,457	57,963	58,481	58,726
$3\frac{1}{2}$	67,347	$3\frac{1}{4}$	53,038	59,389	60,496	61,332	61,986	62,513	62,946	63,304
$3\frac{5}{8}$	72,249	...	...	63,993	65,143	66,006	66,688	67,234	67,684	68,060
$3\frac{3}{4}$	77,315	3	60,722	68,768	69,960	70,854	71,560	72,125	72,592	72,986
$3\frac{7}{8}$	82,556	...	...	73,713	74,950	75,874	76,605	77,189	77,672	78,069
4	87,969	3	70,192	78,834	80,111	81,067	81,823	82,425	82,924	83,340

Permissible Working Loads of Steel Stays with a Stress of 9,000 lbs. per Square Inch.

Outside Diameters	Working Loads for Plus Threads	Whitworth Threads		Number of Threads per Inch						
		Number of Threads		6	7	8	9	10	11	12
			Working Loads of Screwed Stays at 9,000 lbs. per Square Inch							
Inches	Lbs.		Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.	Lbs.
$\frac{3}{4}$	3,976	10	2,740	...	...	...	...	2,734	2,839	2,925
$\frac{7}{8}$	5,412	9	3,795	...	...	...	3,795	3,944	4,069	4,172
1	7,069	8	4,988	...	...	4,988	5,201	5,375	5,520	5,641
$1\frac{1}{8}$	8,946	7	6,275	...	6,275	6,581	6,827	7,026	7,192	7,330
$1\frac{1}{4}$	11,045	7	8,051	...	8,051	8,398	8,675	8,898	9,085	9,240
$1\frac{3}{8}$	13,365	6	9,539	9,539	10,047	10,435	10,742	10,992	11,199	11,371
$1\frac{1}{2}$	15,904	6	11,703	11.703	12,264	12,692	13,032	13,306	13,533	13,723
$1\frac{5}{8}$	18,662	5	13,247	14,087	14,702	15,171	15,541	15,841	16,089	16,295
$1\frac{3}{4}$	21,644	5	15,777	16,692	17,361	17,870	18,273	18,596	18,866	19,089
$1\frac{7}{8}$	24,845	$4\frac{1}{2}$	17,884	19,518	20,241	20,790	21,225	21,573	21,863	22,103
2	28,275	$4\frac{1}{2}$	20,805	22,568	23,341	23,932	24,397	24,772	25,081	25,338
$2\frac{1}{8}$	31,916	$4\frac{1}{2}$	23,942	25,833	26,663	27,293	27,790	28,190	28,520	28,795
$2\frac{1}{4}$	35,782	4	26,330	29,322	30,206	30,874	31,405	31,829	32,180	32,471
$2\frac{3}{8}$	39,871	4	29,850	33,032	33,970	34,680	35,239	35,689	36,006	36,369
$2\frac{1}{2}$	44,177	4	33,590	36,962	37,955	38,704	39,295	39,770	40,163	40,488
$2\frac{5}{8}$	48,707	4	37,560	41,112	42,160	42,951	43,573	44,073	44,487	44,828
$2\frac{3}{4}$	53,457	4	41,739	45,485	46,586	47,417	48,067	48,595	49,031	49,389
$2\frac{7}{8}$	58,427	$3\frac{1}{2}$	44,506	50,079	51,233	52,105	52,780	53,340	53,794	54,169
3	63,617	$3\frac{1}{2}$	49,049	54,893	56,101	57,003	57,729	58,303	58,781	59,171
$3\frac{1}{8}$	69,029	...	53,801	59,928	51,190	62,143	62,889	63,490	63,987	64,395
$3\frac{1}{4}$	74,660	$3\frac{1}{4}$	57,657	65,182	66,498	67,492	68,271	68,896	69,415	69,842
$3\frac{3}{8}$	80,516	...	62,814	70,660	72,030	73,062	73,874	74,524	75,061	75,505
$3\frac{1}{2}$	85,689	$3\frac{1}{4}$	68,192	76,353	77,791	78,855	79,700	80,374	80,930	81,391
$3\frac{5}{8}$	92,883	...	73,793	82,276	83,755	84,865	85,742	86,444	87,023	87,493
$3\frac{3}{4}$	99,398	3	78,171	88,416	89,948	91,098	92,006	92,732	93,333	93,824
$3\frac{7}{8}$	106,134	...	84,056	94,774	96,364	97,550	98,490	99,243	99,864	100,374
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